POWER PLANT TESTING

Published by the

McGraw-Hill Book Company

New York

Successors to the Book Departments of the

McGraw Publishing Company   Hill Publishing Company

Publishers of Books for

Electrical World   The Engineering and Mining Journal

Engineering Record   American Machinist

Electric Railway Journal   Coal Age

Metallurgical and Chemical Engineering   Power

POWER PLANT TESTING

A MANUAL OF TESTING ENGINES, TURBINES, BOILERS, PUMPS, REFRIGERATING MACHINERY, FANS, FUELS, LUBRICANTS, MATERIALS OF CONSTRUCTION, ETC.

BY

JAMES AMBROSE MOYER, S.B., A.M.

Member American Society of Mechanical Engineers, Mitglied des Vereines deutscher Ingenieure, Membre titulaire Association Internationale du Froid, Member of the Franklin Institute, Society of Automobile Engineers, American Institute of Electrical Engineers, etc.

Professor of Mechanical Engineering in The Pennsylvania State College and in charge of Pennsylvania Engineering Experiment Station, formerly Engineer, Steam Turbine Department, General Electric Company, and Engineer, Westinghouse, Church, Kerr & Company

SECOND EDITION

REWRITTEN, ENLARGED AND ENTIRELY RESET

McGRAW-HILL BOOK COMPANY, INC.

239 WEST 39TH STREET, NEW YORK

6 BOUVERIE STREET, LONDON, E. C.

1913

Stanhope Press
F. H. GILSON COMPANY
BOSTON, U.S.A.

PREFACE TO FIRST EDITION

IN the preparation of this book the object in view has been primarily to give in a small volume, somewhat in detail, the generally approved methods of testing engines, turbines, boilers and the auxiliary machinery usually found in power plants, as well as to present more or less complete descriptions of the various kinds of apparatus used and the calibrations required for accurate testing. In addition to this subject-matter, chapters have been prepared on the testing of fuels, refrigerating and hydraulic machinery, as well as on the proper methods and the machinery to be used in making tests of the strength of the materials commonly used in the construction of buildings.

As a book for students in laboratory courses it is intended particularly for use in large classes in which at the beginning of the laboratory periods it is necessary to begin at the same time a number of different experiments and tests. On this account care has been taken to state as clearly as possible the descriptions of the apparatus to be used and the precautions to be observed to secure accuracy in the results. Students should be expected, however, to rely to some extent on their own initiative.

In most respects the book is probably complete enough in descriptive matter and in general instructions so that very little lecture-room work is needed for at least elementary courses. It is the author's opinion that students in experimental engineering laboratories should not receive a great deal of assistance in planning and conducting tests. Sometime they must learn to be resourceful and independent of the "school" type of instruction and obviously the sooner this is appreciated by both instructors and students the greater will be the benefits. At least for very small classes the better plan is the one advocated years ago by a famous educator, that students working in laboratories when assigned the work of testing a machine, a new type of aeroplane engine, for example, should have very simple instructions such as: "Make tests of this new type of engine, find out what you can about it and report your results." It is to be hoped that the particular method of teaching in laboratories, known familiarly as "feeding with a spoon," has disappeared in present-day instruction in technical schools and colleges.

Quite a large part of the training required for one to become accurate and reliable in the work of observing and interpreting the results of

tests of machinery consists in becoming familiar with the details of the adjustment and calibration of the various instruments, so that they may be used intelligently.

Although in the arrangement of the chapters the use of the book by students was given the most careful consideration, yet as a whole the needs of the "practical" man were not lost sight of, and it is hoped that the author's experience when working with this group of readers in testing both large and small power plants has helped to make the book interesting and helpful to them. The book is intended to be also a manual giving useful information in a more or less limited way to those professional engineers having the advantages of a technical training, but who are not thoroughly familiar with the most up-to-date methods of testing.

In many cases not nearly all the results that should be calculated to make up a complete report are mentioned. It is the opinion of the author that in a text-book it is desirable that no more than very general instruction should be given regarding the conduct of the test, the quantities to be calculated, and the form and tabulations expected in a report. Such details should be left in the hands of the instructor. Because the size of the book was limited it was necessary to omit explanations of the methods of calculating many interesting and more or less applicable results from tests. In the more extended courses it is believed that the instructors can readily fill in these omissions.

The author is particularly indebted in the preparation of this book to Dean M. E. Cooley and Professor J. R. Allen, of the University of Michigan; Professor L. S. Marks, of Harvard University; Professor H. W. Spangler, of the University of Pennsylvania; Dean W. F. M. Goss, of the University of Illinois; Professor C. H. Peabody, of the Massachusetts Institute of Technology; Professor L. V. Ludy, of Purdue University; Professor A. M. Greene, of Rensselaer Polytechnic Institute; Professor C. C. Lorentzen, of New York University; Professor E. J. Fermier, of the Mechanical and Agricultural College of Texas; Professor E. A. Fessenden, of the University of Missouri; Professor F. H. Sibley, of the University of Alabama; Dr. C. P. Steinmetz and Mr. Richard H. Rice, of the General Electric Company; Mr. H. R. Kent, Vice-president, Westinghouse, Church, Kerr & Company; Mr. J. R. Bibbins, of the Arnold Company; Mr. R. A. Smart, of the Westinghouse Machine Company; Mr. St. John Chilton, of the Allis-Chalmers Company; and Mr. G. E. Wallis, New York City.

J. A. Moyer.

Ann Arbor, Michigan.
August, 1911.

PREFACE TO SECOND EDITION

On account of the very great extension of experimental work in engineering laboratories, and the great demand for a suitable manual of small size and clearly written for the use of students, this book has been largely rewritten and a number of new sections have been added to make it more adaptable to the large number of engineering schools using it as a text.

The latest revisions (1912) of the standard codes adopted by the Power Test Committee of the American Society of Mechanical Engineers have been incorporated practically without abridgment to make the book thoroughly up-to-date.

Outlines of a series of tests arranged to suit as nearly as possible a large number of American technical schools have been added to assist both the instructor and the student. By the judicious use of these outlines at least fifty per cent of the time of an instructor can be saved; the general efficiency of the courses will be improved; and the predominating idea underlying the original conception of this book as stated in the third paragraph of the first preface is more nearly realized.

The most obvious weakness of laboratory instruction to be observed, particularly when large classes are the rule, is that students have assimilated with thoroughness very little of what has been taught. The principal difficulty is that the students have been permitted to depend too much on laboratory notes prepared by the instructors, which are complete enough for the particular test in hand but are not sufficiently general in the discussions. It is the general conclusion of many who have investigated, that the only way out of this difficulty is to require written recitations, say once every week or every other week for about a half hour to review the general subject matter and the exact details of the methods used in making the tests.

It is a good plan also for the instructor to require students to make approximate calculations of the principal results of each test, under his personal supervision, before they report for making the accurate calculations going into their reports. In this connection the instructor should feel it his duty to be certain that the students understand the methods of obtaining the important results before a laboratory test is started. There is no better place than an engineering laboratory to impress the importance of rough calculating, plotting, and checking of results than

such work done while a test is in progress. One-half of the data generally collected by undergraduates doing "thesis work" is practically valueless because the importance of calculating roughly for final and important results is not properly appreciated. If such work is not to be done well it might better not be done at all.

In the preparation of this edition the author is particularly indebted to Dean F. P. Anderson, Univ. of Ky.; Mr. G. H. Barrus, Boston, Mass.; Prof. J. P. Calderwood, Pa. State Col.; Prof. F. E. Cardullo, New Hampshire Col.; Prof. R. C. Carpenter, Miss. A. & M. Col.; Prof. A. G. Christie, Univ. of Wis.; Prof. A. W. Cole, Purdue Univ.; Prof. J. E. Emswiler and Prof. C. H. Fessenden, Univ. of Mich.; Prof. S. H. Graf, Oregon State Col.; President I. N. Hollis, Worcester Poly. Inst.; Prof. A. C. Jewett, Univ. of Maine; Prof. W. H. Kavanaugh, Univ. of Minn.; Prof. W. H. Kenerson, Brown Univ.; Prof. E. W. Kerr, La. State Univ.; Prof. E. H. Lockwood, Yale Univ.; Mr. F. R. Low, Editor of *Power;* Prof. L. S. Marks, Harvard Univ.; Prof. R. B. Otis, Col. Sch. of Mines; Prof. A. A. Potter, Kans. State Col.; Prof. F. L. Pryor, Stevens Inst. of Tech.; Mr. W. W. F. Pullen, London, Eng.; Prof. C. R. Richards, Univ. of Ill.; Prof. R. Royds, Glasgow (Scotland) Tech. Col.; Sir H. R. Sankey, Ealing, Eng.; Prof., J. C. Smallwood, Syracuse Univ.; Prof. C. J. Tilden, Johns Hopkins Univ.; Prof. A. F. Walker, Univ. of Kansas; Prof. H. C. Weaver, Univ. of Texas; Prof. A. C. Wescott, Univ. of Mo.; and Prof. G. S. Wilson, Univ. of Washington.

J. A. Moyer.

State College, Pennsylvania,
August, 1913.

INTRODUCTION

TESTS of the machinery in a power plant are usually made to determine the capacity and efficiencies of its various units when operating under certain definite conditions. In recent engineering practice manufacturers and contractors are generally required to make certain estimates and guarantees of the capacity and efficiency of the various kinds of machinery supplied. This is exactly equivalent, in other words, to agreeing to provide for doing a given unit of work under specified conditions at a definite cost. The purchaser, on the other hand, for his protection, finds it necessary to determine from the results of reliable tests whether the " guarantees " can be obtained. Obviously, then, the importance of knowing how to make careful and reliable tests, of which the results will not be questioned, cannot be overestimated.

Tests of power plants as a whole are also necessary and should be made from time to time in order to determine what results can be obtained from an economic viewpoint when operating under the existing conditions; and also for finding out what saving can be obtained by changes in the operating conditions or by the installation of more efficient auxiliary machinery. From the viewpoint of determining whether or not it is economical to replace old equipment with new, tests of old installations are relatively more important than those of newer ones, because usually it is out of the question to relegate practically new machinery to the scrap-heap. The greatest importance of such tests is due, however, to the fact that they show how nearly the existing conditions of operation conform to those of standard engineering practice, and to those obtained in other plants operating with the greatest success.

Practice tests in the laboratory are intended to show to students by **actual experience** the best methods for investigating the problems arising in the operation of plants, how to work out in a practical way the doubtful points in designing and constructing machinery, and, above all, to think accurately and systematically in such matters.

Procedure for making accurate tests may be stated as follows:

1. Procuring a suitable standard testing equipment. Any instruments and apparatus not well known to engineers generally and which are of doubtful accuracy or sensitiveness should always be avoided. Remember that a single element of uncertainty may vitiate the acceptance of the results of a test of otherwise undoubted accuracy.

2. Careful calibrating of instruments before a test, so that the greatest possible errors of the tests are definitely known and that proper allowance can be made in the results.

3. Systematic recording of observations.

4. Recalibrating of instruments after a test to determine whether there have been any changes in their accuracy.

5. Preparing of a report embodying data, results and conclusions.

6. Tabulating and plotting on cross-section paper the important results. This plotting is not only for the purpose of showing the results graphically, but also for the purpose of providing a check or a method of eliminating errors in observations or in calculations. The skill of an engineer in testing is shown, more than in any other way, by his **ability to check results.** If after applying various checks, usually by means of plotted curves, the results for varying conditions are found to agree, the engineer is able to tell definitely whether or not his tests are reliable.

CONTENTS

POWER PLANT TESTING

CHAPTER I

MEASUREMENT OF PRESSURE

The simplest instrument used for measuring pressure is a glass tube bent into the shape of the letter U, as illustrated in **Fig. 1**. When such a tube, called technically a **manometer** or **U-tube,** is partly filled with a liquid, usually water or mercury, and is connected at **A** by means of tubing to the vessel in which the pressure is desired, there will be observed a difference in the level of the liquid corresponding to the pressure. If the end of the tube at **B** is open to the atmosphere, then the difference in the level of the liquid in the two legs measured in inches, multiplied by the weight of a cubic inch of the liquid in pounds, gives the difference in pressure in pounds per square inch between that in the vessel and atmospheric pressure. When the level in the leg **B** is higher than in **A** then the pressure measured is greater than atmospheric and is called **gage pressure** to distinguish it from the other condition when the level in the leg **A** is higher than in **B**; that is, when the pressure is less than atmospheric. In the latter case we speak of **vacuum** or negative pressure.

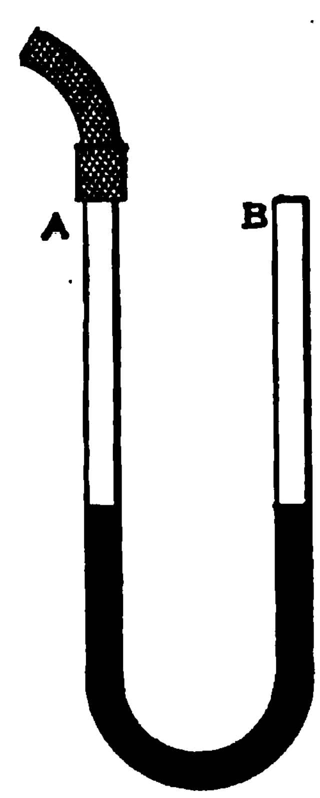


Fig. 1. — A U-tube for measuring Pressures.

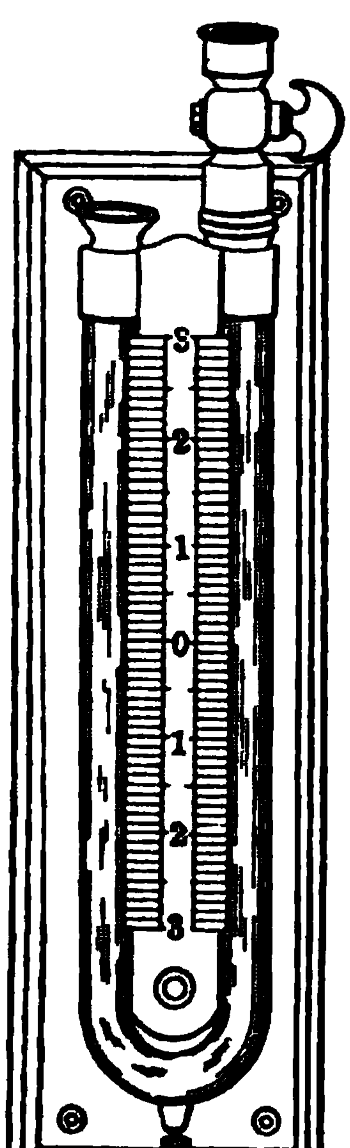


Fig. 2. — Manometer or U-tube with Graduated Scale.

As such instruments are usually constructed, a scale suitably graduated for measuring the difference between the levels of the liquid in the tube is placed between the two legs, as shown in **Fig. 2**. Still another type is illustrated in **Fig. 3**. In a manometer of this kind one leg can be made very short if it is correspondingly large in diameter. If the scale is adjusted so that the level in the short leg is at the zero of the scale, then the level in the long leg will indicate directly inches of pressure or of vacuum

as the case may be. A typical vacuum gage of the same kind is illustrated in **Fig. 4.** The end of the tube corresponding to the short leg in **Fig. 3** is shown at **A.** When manometers are to be used for pressure or vacuum measurements of **steam,** a condensation trap (**B, Fig. 5**) is often employed to prevent the passage of steam into the glass tube in which it would form a water column on the top of the mercury for which a correction[1] would have to be made. To be effective the condensation trap

FIG. 3. FIG. 4. FIG. 5.

Typical Mercury Vacuum Gages.

B must always be partly filled with water. It must not however be allowed to become completely filled so as to discharge water through the pipe **D** joining the trap with the glass vacuum tube. A glass tube **E,** called technically a **water gage,** shows the level of the water in the trap

[1] Correction for water on the top of a mercury column is most conveniently made by dividing the length of the water column by the specific gravity of mercury (13.6) and adding this equivalent length to the mercury column on which the water rests.

and should always be kept clean. Cock **C** is provided for draining off excess of condensation.

The graduated scales on vacuum gages, like **Figs.** 3, 4, and 5, must be adjustable so that the zero can be made to coincide exactly with the level of the mercury when both legs are open to atmospheric pressure. Scales are usually arranged to be raised and lowered by turning a milled knob, like **K** in **Fig.** 5, which is connected by a screw thread to the scale. Vacuum gages, like **Fig.** 3, without ready means for adjustment cannot be arranged to indicate accurately vacuums that are varying, because

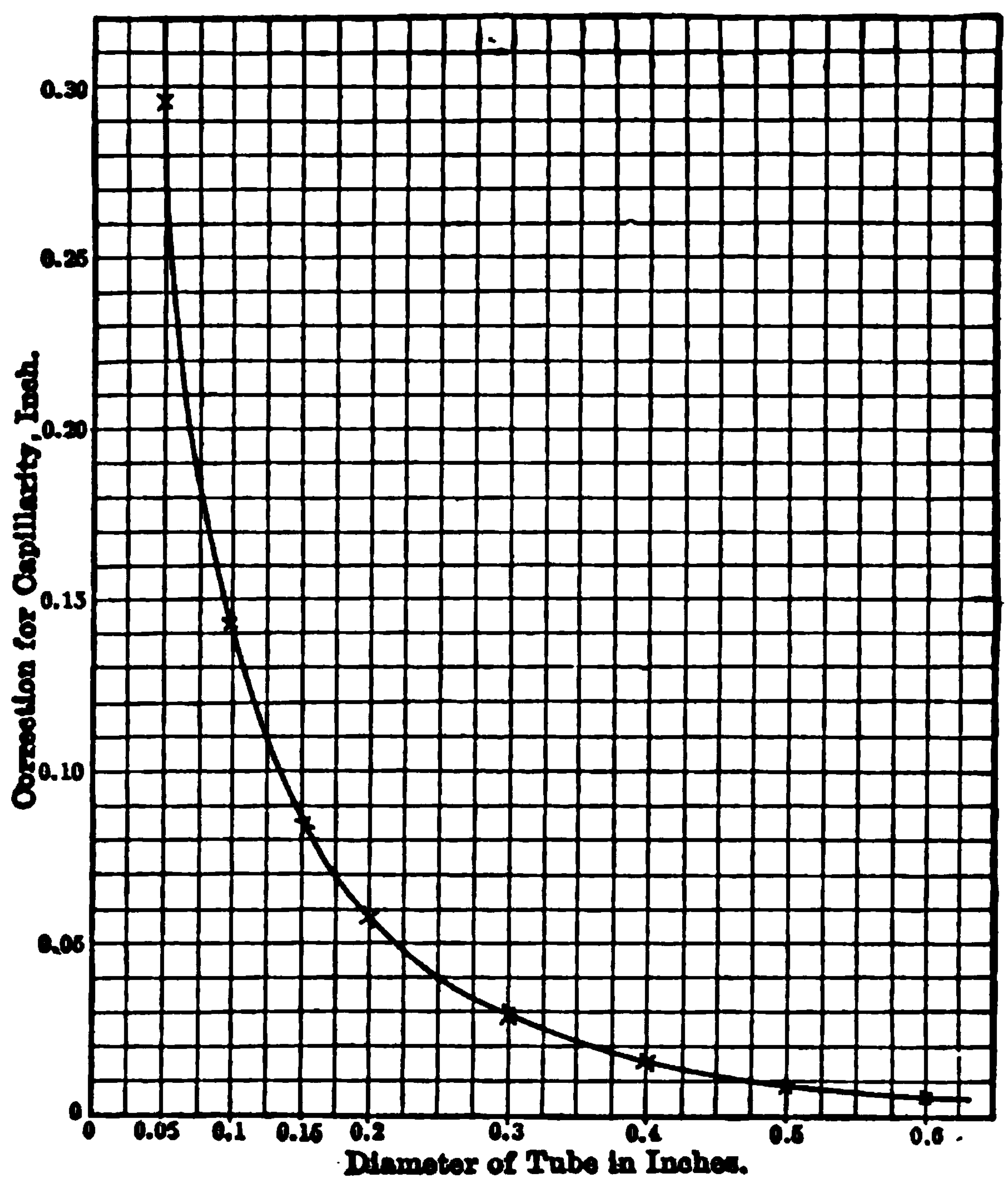


FIG. 6. — Curve of Capillarity Corrections for Mercury Columns.

the measure of the vacuum is the difference in level between the columns in the two legs. Unless the area of the reservoir is very large, in all vacuum manometers either the scale must be shifted or the level in the reservoir changed for varying magnitudes of vacuum.

Manometers or U-tubes of very small diameter when filled with mercury may be affected by capillarity to such an extent that in order to obtain the true height corresponding to the pressure, a correction must be added. It is not at all unusual to find manometers used for vacuum gages to be comparatively small in diameter, and unless the graduations of the scale have been corrected for the error due to capillarity the proper

allowances must be made for all observations. **Fig. 6** shows by a curve the values of this correction as determined by Pullen for mercury columns of various diameters.

Mercury columns should be read at the top of the meniscus and water columns should be read at the bottom. In this way, except in very small tubes, the errors due to capillarity may be regarded as negligible. If the graduated scale is made to lap over nearly half the width of the tube on each side, as illustrated in **Fig. 2**, and all the observations are taken at the meniscus in the middle of the tube, a remarkable degree of accuracy is obtainable.

Mercury barometers are instruments designed to accurately measure atmospheric pressure. The one shown in **Fig. 7** consists of a glass tube, closed at the top and enclosed in a metal casing **T**. The open end of the glass tube at the bottom dips into a glass cup **C** well below the level of the mercury with which it is partly filled. The glass cup is closed at the bottom by a leather bag resting on a movable disk. By means of the screw **S** the level of the mercury can be raised or lowered until the surface of its meniscus just touches the tip of a fixed ivory point. Atmospheric pressure acts upon the mercury in the cup so that the height of the mercury column in the tube is a measure of this pressure. A vernier is generally provided alongside the scale to assist in accurate observations of the height, which is measured to the top of the meniscus. For every observation the level of the mercury in the cup **C** must be adjusted accurately to the tip of the ivory point. The equipment of a modern power plant is incomplete without a reliable and accurate barometer.

Fig. 7. — Mercury Barometer.

Standard barometric pressure for comparison is usually taken as 30 inches of mercury at a temperature of 32 degrees Fahrenheit. To correct a barometric reading and obtain the equivalent height at 32 degrees corrections must be made for both the cubical expansion of mercury (coefficient = .000100) and for the linear expansion of the scale (coefficient of brass = .000011). If **H** is the barometric reading in inches at t degrees Fahrenheit, then the equivalent height corrected for temperature (H_0) is

$$H_0 = H \left\{ 1 - \frac{(.000100 - .000011)(t - 32)}{1 + .000100\,(t - 32)} \right\} \quad . \quad . \quad . \quad . \quad (1)$$

This equation is equivalent in English units to the standard correction adopted by the International Bureau of Weights and Measures.[1]

Some observers, however, have adopted 62 degrees Fahrenheit instead of 32 as the standard for the **brass scale,** making the correction considerably more complicated. Whether the scale is corrected to 32 or 62 degrees actually changes the results very little.

Vacuum measured with a mercury manometer must be corrected to standard temperature conditions in the same way as barometer readings are corrected. Frequently wooden scales are used on vacuum manometers. If the scale has been cut so that the length is along the grain the expansion is very small (see Table II in the Appendix) and the equation above can be simplified for most practical purposes to

$$H_0 = H - .0001\,(t - 32) \qquad (1')$$

when 32 degrees Fahrenheit is the standard for comparison.

It often happens that there are tens and even hundreds of feet difference in elevation between the location of the barometer and that of the vacuum gages. It then becomes necessary to correct the reading of the vacuum gages to the elevation of the barometer. The correction is about 0.11 inch[2] per 100 feet, which is to be added to the vacuum reading when the vacuum gage is elevated above the barometer, and *vice versa.* This correction for elevation is often serviceable for determining the approximate barometer reading at a place where no barometer is available, by applying the corrections to the barometer observation at the nearest station of the U. S. Weather Bureau. Observations reported by the Weather Bureau are at sea-level and 32 degrees Fahrenheit. To use these data it is therefore necessary to know the elevation above sea-level of the place where the test is being made, and all observations made with mercury columns must be reduced to the equivalent at 32 degrees Fahrenheit to obtain correct absolute pressures. (See Report of Power Test Committee of American Society of Mechanical Engineers in *Journal of the American Society of Mechanical Engineers*, Nov., 1912, page 1696.)

Obviously, temperature and elevation corrections are avoided when the barometer and the vacuum manometer are hung very near each other. To correct observations of vacuum to equivalent vacuum compared to (or " referred to ") 30 inch barometer, the difference between the corrected barometer and 30 inches is added to the observed vacuum when the barometer is less than 30 inches, and is subtracted when greater than 30 inches. This is the method most generally used.

[1] Broch, *Bulletin of International Bureau of Weights and Measures* (1887).

[2] A cubic foot of air at average atmospheric temperatures weighs about .078 pound or $(.078 \times 100) \div 144 = .0542$ pound per square inch for 100 feet of elevation, which is equivalent to $.0542 \div 491$ or .11 inch mercury per 100 feet.

Aneroid Barometer. Atmospheric pressure is sometimes measured by a mechanical device called an aneroid. It is simply a delicate pressure gage which is more conveniently portable than a mercury barometer. On this account aneroids are frequently used in power plant testing and for determining elevations. They must be frequently calibrated by comparison with a mercury barometer. Most engineers place more reliance on aneroids than their accuracy permits.

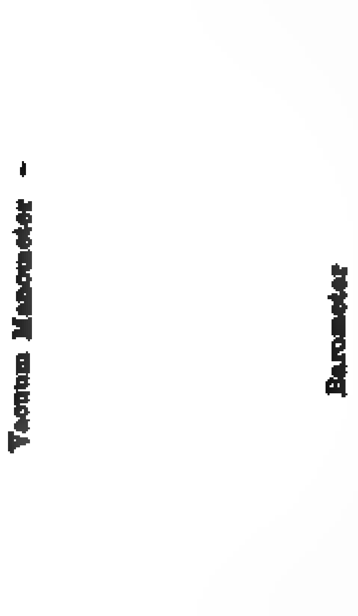


FIG. 8. — Simplest Form of Absolute Pressure Gage.

Absolute Pressure Gages. Simplest forms of instruments for indicating directly absolute pressure consist of a mercury barometer and a vacuum gage placed side by side, with a sliding scale between them, as illustrated in **Fig. 8.** When the zero is adjusted so as to be exactly opposite the top of the mercury column in the barometer, the reading on the scale opposite the level in the vacuum gage is the absolute pressure. Obviously the scale can be graduated to indicate inches of mercury, pounds per square inch, or any similar units of absolute pressure. A simpler commercial form of absolute pressure gage is shown in **Fig. 9,** designed to show directly without adjustment the absolute pressure. Essentially it is a barometer with only a very short tube provided for the mercury column. Instead, however, of having the mercury cup or reservoir at the bottom open to atmospheric pressure as in the ordinary barometer, the cup **C,** in this case, is sealed except for the pipe **P,** which is to be connected to the chamber in which the absolute pressure is to be measured. Before filling the instrument with mercury, the tube is exhausted to a practically perfect vacuum as for a regular-size barometer column. When the pipe **P** is open to atmospheric pressure, the mercury column will be forced all the way to the top of the capillary tube; but when this pipe is connected to a vacuum chamber, the mercury column will gradually fall. If the instrument is well made it should

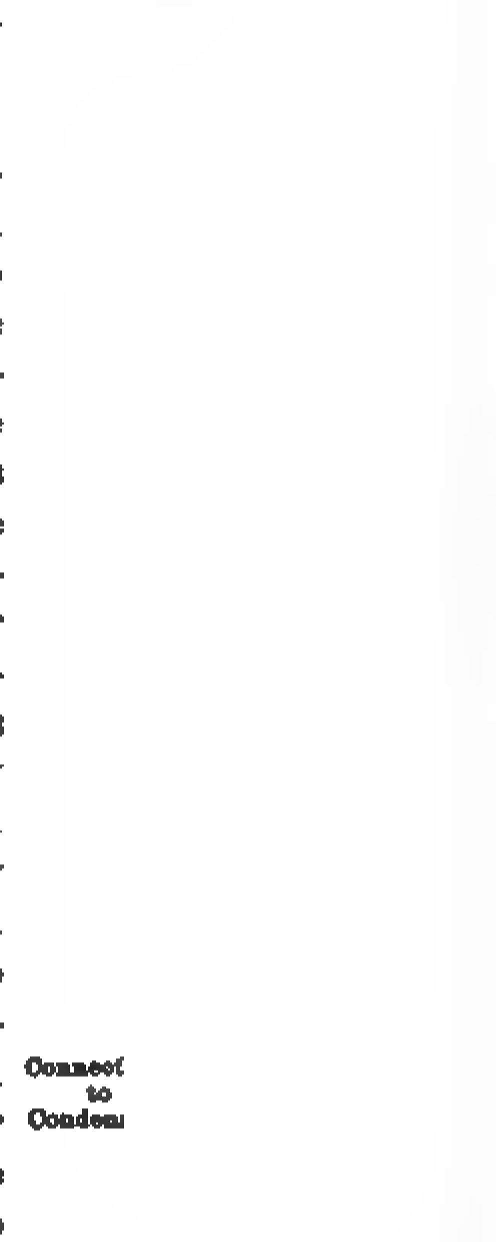


FIG. 9. — Commercial Form of Absolute Pressure Gage.

be fairly accurate for the range between four pounds and one-half pound per square inch absolute pressure. Graduations on the left side are inches of mercury and on the right, pounds per square inch, absolute pressure.

Conversion of Pressures. It is frequently necessary to reduce pressures in inches of mercury or of water to the equivalent in pounds per square inch. Since the weight of a cubic inch of mercury at 70 degrees Fahrenheit is .4906 pound and of water at the same temperature is .0360 pound, pressures in inches of mercury at the usual "room" temperatures can be reduced to pounds per square inch by multiplying by .491 or by dividing by 2.035, and similarly inches of water can be converted to pounds per square inch by multiplying by .0360 or by dividing by 27.78. Centimeters of mercury are reduced to pounds per square inch by multiplying by .1903.

Kilograms per square centimeter are reduced to pounds per square inch by multiplying the kilograms per square centimeter by 14.223 or by dividing by .0703. Grams divided by 28.35 are ounces avoirdupois; or one gram is approximately $\frac{1}{30}$ ounce.

General metric conversion tables are given in Table III in the Appendix.

A cubic foot of water at 70 degrees Fahrenheit weighs 62.3 pounds and at 30 degrees Fahrenheit, 62.4 pounds. At ordinary "room" temperature the pressure due to 2.31 feet of water is equivalent to one pound per square inch.[1]

Tubes used as mercury manometers must be cleaned from time to time by washing the inside surface with nitric acid and afterward thoroughly cleansing them with water. Mercury used in manometers should be free from impurities. Usual impurities can generally be removed by filtering through a clean cloth of close texture or a very thin chamois leather. Air can be removed by boiling, but by far the best method for cleaning mercury is by means of a mercury still. Unfortunately an apparatus of this kind is not available in most engineering laboratories.

Pressure Gages. The large size necessary, however, for manometers or U-tubes, even if filled with the heaviest liquids, makes their use unsuitable except for comparatively low pressures. Instruments more desirable for high pressures are made by the application of some kind of elastic material designed to produce a uniform deformation for variations of pressure. By connecting a suitable auxiliary mechanism to the elastic element it can be made to move a needle to indicate on a graduated dial the degree of pressure. The most common form of such devices is a

[1] The unit pressure of one pound per square inch is equivalent also to that due to a column of air of uniform density, of which the vertical height in feet is approximately 144.0 divided by the weight of a cubic foot of air at the temperature, pressure and humidity as observed. Tables of the weight of air are given on page 181.

hollow brass or steel tube bent into the shape of an arc of a circle. It is a well-known principle that when a straight piece of tubing is bent into this shape the sides come nearer together, making the section of the tube a very much flattened oval.[1] A tube of this kind is illustrated in **Fig. 10**, showing also in the right-hand corner a transverse section. If one end of such a tube is closed and fluid pressure is applied to the inside, the parallel sides, as at **A** and **B**, tend to separate and consequently there is a tendency for the radius of curvature of the tube to become larger, thus moving the end at **E** toward **F**. By connecting a suitable mechanism to **E**, the degree of pressure can be indicated. Instruments of this kind are called **Bourdon gages.**

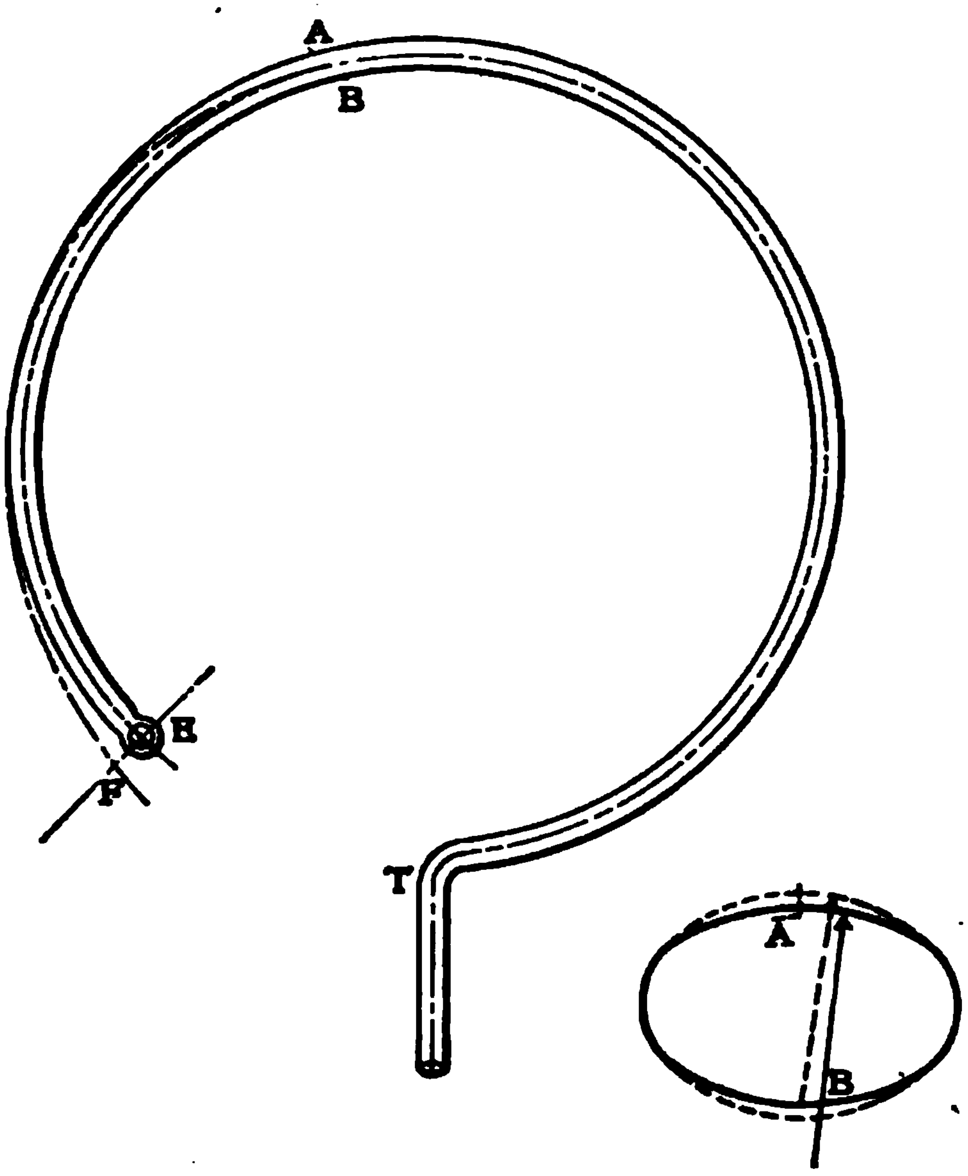


Fig. 10. — A Typical Bourdon Tube.

Fig. 11 shows one of the simplest forms of such gages used in power plants to indicate the pressures. It consists essentially of the curved tube **T** of oval cross-section closed at one end. This end is connected by means of suitable levers to a rack **R**, engaging with a small pinion **N** on the same shaft with the pointer or needle **P**. Pressure applied to the tube **T** causes the rack and pinion to move over a dial (**Fig. 12**) graduated or marked to indicate pressures in standard units as, for example, pounds per square inch (English system) or kilograms

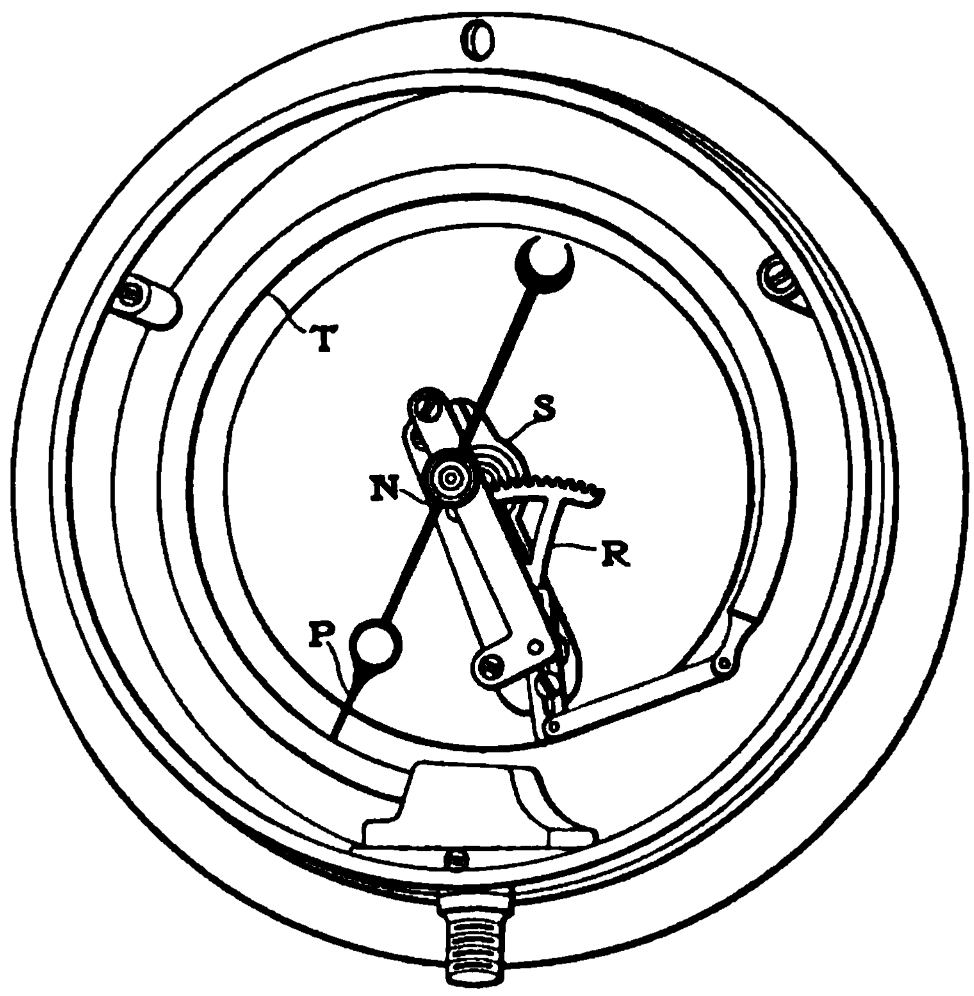


Fig. 11. — Mechanism of Bourdon Tube Gage.

[1] For a theoretical discussion of this principle in detail see *Theorie der Röhrenfeder Manometern* in *Zeitschrift des Vereines deutscher Ingenieure*, Oct. 29, 1910, pages 1865–73.

per square centimeter (Metric system). Gages are usually most sensitive and accurate from about one-half to two-thirds of the maximum graduation. The gage shown in **Fig. 12** is most suitable for use between 130 and 175 pounds per square inch.

Fig. 13 shows a form of Bourdon gage in which the amount of vibration of the needle due to the jarring that occurs in locomotive and other portable services has been reduced to a minimum by supporting the pressure tube in the middle instead of at its end as in **Fig. 11**. This form of tube has also advantages for use in gages exposed to temperatures below freezing, since the arms can be drained of water, while the other form will usually hold the water that has entered.

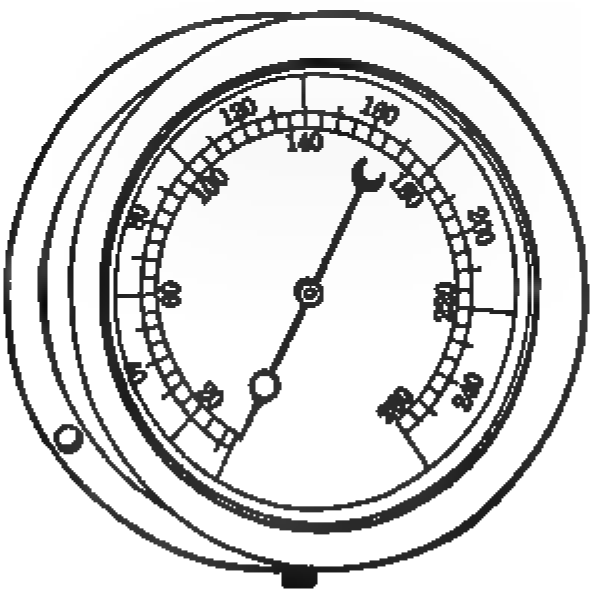


FIG. 12. — The Dial of a Pressure Gage. FIG. 13. — A Modified Bourdon Gage.

In Bourdon gages any lost motion of the parts is taken up by the hair spring attached to the spindle carrying the pointer. But the use of a Bourdon pressure gage under conditions of constant vibration due to either moving over roads and fields on a tractor or to the fluctuation of pressure in a pipe due to the sharp cut-off of an engine will rapidly wear off the teeth of the rack **R** and of the pinion **N** (**Fig. 11**), so that it is not unusual to find some gages very inaccurate in the portion of the scale most used.

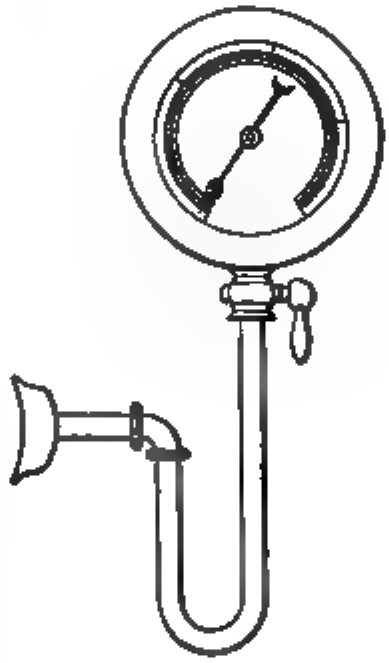


FIG. 14.

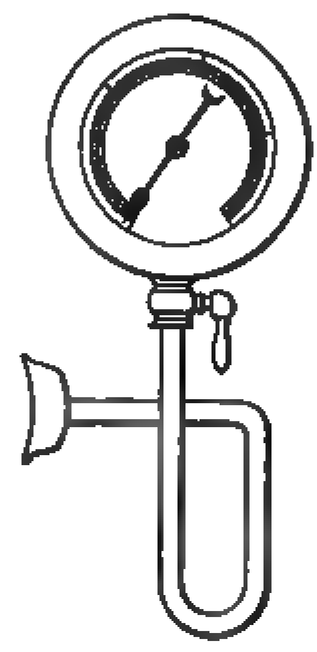
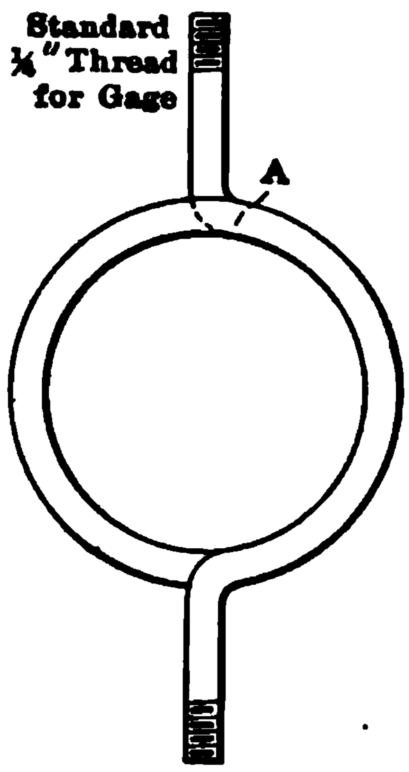


FIG. 15

Examples of "Goose-neck" Steam Siphons.

Gages to be used to determine the pressure of ammonia have the oval tube made of steel instead of brass because the latter material deteriorates rapidly in the presence of ammonia.

Bourdon gages may be used for indicating the pressures of either liquids, steam or gases without observing special precautions if the temperature is never much over 150 degrees Fahrenheit. If, however, the elastic tube in the gage is heated above this limit it is likely to lose some of its temper. When used for steam pressure, therefore, **some form of siphon or water-seal must always be used** to prevent steam from entering the gage. The forms of siphon shown in **Figs. 14** and **15** are preferred for accurate measurements, but the types used most commonly are illustrated in **Figs. 16–18.** Ordinarily brass pipe is preferred to iron, because of its better conductivity. This suggestion is particularly important for superheated steam. In the siphons shown in **Figs. 16** and **17** there is always a possibility, however, that air carried in the steam may be entrapped at **A** where it forms a cushion, preventing the gage from indicating the true variation in pressure. **Fig. 18** represents a very common device for putting a water-seal between a steam pipe and a gage. Although not considered as effective as some of the other types shown, it has the advantage of being more compact.



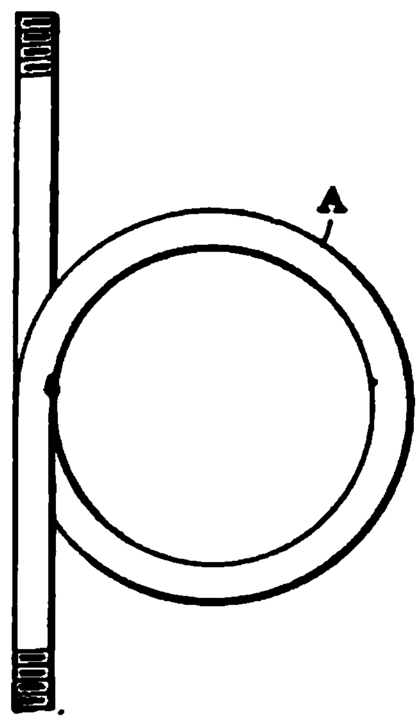


FIG. 16. FIG. 17.
Examples of Ring Steam Siphons.

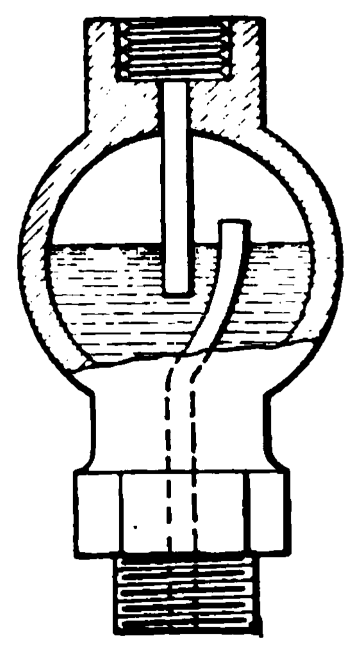


FIG. 18. — Simple Water-seal Steam Siphon.

Adjustments. The ratio of motion of the pointer with respect to that of the tube can be adjusted in most Bourdon gages by sliding a set-screw in a slot in the short arm of the rack-lever. In the gage illustrated in **Fig. 11** when the short arm of the rack-lever is made longer by adjusting the set-screw, the movement of the rack and also of the pointer is reduced for a given deflection of the tube.

Sometimes when used carelessly, especially when subjected to pressures beyond the scale on the dial, the tube of the gage takes a permanent "set"; or, in other words, it does not spring back to its original position, and the pointer does not come back to the zero mark. In such exigencies and also for adjustment after calibration the needle can be forced off from its spindle — preferably by the use of a clamp or "needle-jack"

made by gage manufacturers specially for this service — and then set again in position where it should be.

Another kind of gage in which there is a metallic disk or diaphragm instead of a bent tube for actuating the indicating device is sometimes used. One of this type is well illustrated in **Fig. 19.** It consists of a **corrugated diaphragm** clamped around its edge by the flanges of an encircling chamber. Pressure applied on the lower side of the diaphragm deflects it upward, the amount of this upward movement being proportional to the pressure. By means of a connecting strut S the movement of the diaphragm is communicated to a rack R connected to a small pinion attached to the spindle of the needle indicating the pressure on the graduated dial.

Since the deflection of the center of the diaphragm is proportional to the pressure and is inversely proportional to the cube of its thickness, a very slight alteration in the thickness of the diaphragm will cause a considerable change in the reading of the gage.

FIG. 19. — A Typical Diaphragm Gage.

Hydrostatic pressure must be taken into account in some places, particularly when a Bourdon or diaphragm gage is used for determining the pressure in a boiler or in a water pipe. For example if the gage is located below the surface of the water in a boiler the gage will read too high, the error being the pressure equivalent to the head of water between the surface in the boiler and the center of the gage. The correction in pounds per square inch is .433 times the head in feet. To make this correction unnecessary the gage, if of the Bourdon type, should be located so that its center will be at about the average water-level in the boiler; and if it is of the diaphragm type, the diaphragm should be at about the average water-level.

The practice of **choking the gage cock** to reduce vibrations of the pointer, unless done very carefully, is objectionable, especially when there is leakage around the joint of the plug in its cock. Under these circumstances the true pressure will be greater than that indicated.

Tapping a gage, preferably on the back, just enough to observably move the pointer, is an essential precaution to take before reading a gage having levers, racks, and springs in its mechanism. This precaution is necessary to be certain that all working parts are moving freely. (See Report of Power Test Committee in *Journal of A.S.M.E.*, Nov., 1912, pages 1695–6.)

Vacuum Gages. For the measurement of vacuum instead of pressure Bourdon gages are very commonly used. The design for a pressure gage is altered only in the arrangement of the levers moving the needle, which for vacuum measurements turn the needle in the same direction as for pressure (clockwise); but in this case, the tube is bent inward or toward the center of the gage instead of outward as for pressure measurements. Vacuum gages are usually graduated to read **inches** of mercury below atmospheric pressure. **Absolute pressure in inches of mercury** is the difference between the barometer and the reading of such vacuum gages.

A vacuum gage is generally used to determine the **tightness of vacuum lines,** condensers, etc. If the line is tight the gage will show no appreciable diminution of vacuum for several hours, but it drops quickly if there are leaks. These are most readily located by bringing a candle flame close to all the possible places of leakage. The flame will be drawn strongly toward the leaks by the current of air which is being drawn into the pipe by the vacuum inside.

Another type known as a **compound gage** is used to indicate either **pressure** or **vacuum** on the same dial. The linkages are adjusted differently from those in the usual types in that the position of the pointer for zero gage pressure would now be at about the point marked 140 in **Fig. 12.** Graduations to the left around the dial would be from zero to thirty inches of mercury (vacuum), and similarly, to the right, from zero to about fifteen pounds per square inch. Such a device permits using a gage with a single Bourdon tube for measuring either vacuum or a considerable range of pressure.

Differential Gages are designed to read directly pressure difference. Most commonly they are U-tube manometers or equivalent devices arranged to have each side (or leg) connected to a source of pressure. The displacement of the liquid columns of the manometer will indicate the difference in pressure. In **Fig. 226** (page 182) the U-tubes marked **a** are good examples of differential manometers or gages. Such gages are very satisfactory for such purposes as measuring the difference in pressure on the two sides of an orifice in steam, air, or water pipes.

Recording Gages. In many modern power plants recording gages are used to give a graphic record on a chart of the pressure or vacuum for 24 hours. The most common type of recording gage is shown in **Fig. 20.**

FIG. 21.—Operating Parts of a Recording Gage with a Helical Tube. (Bristol.)

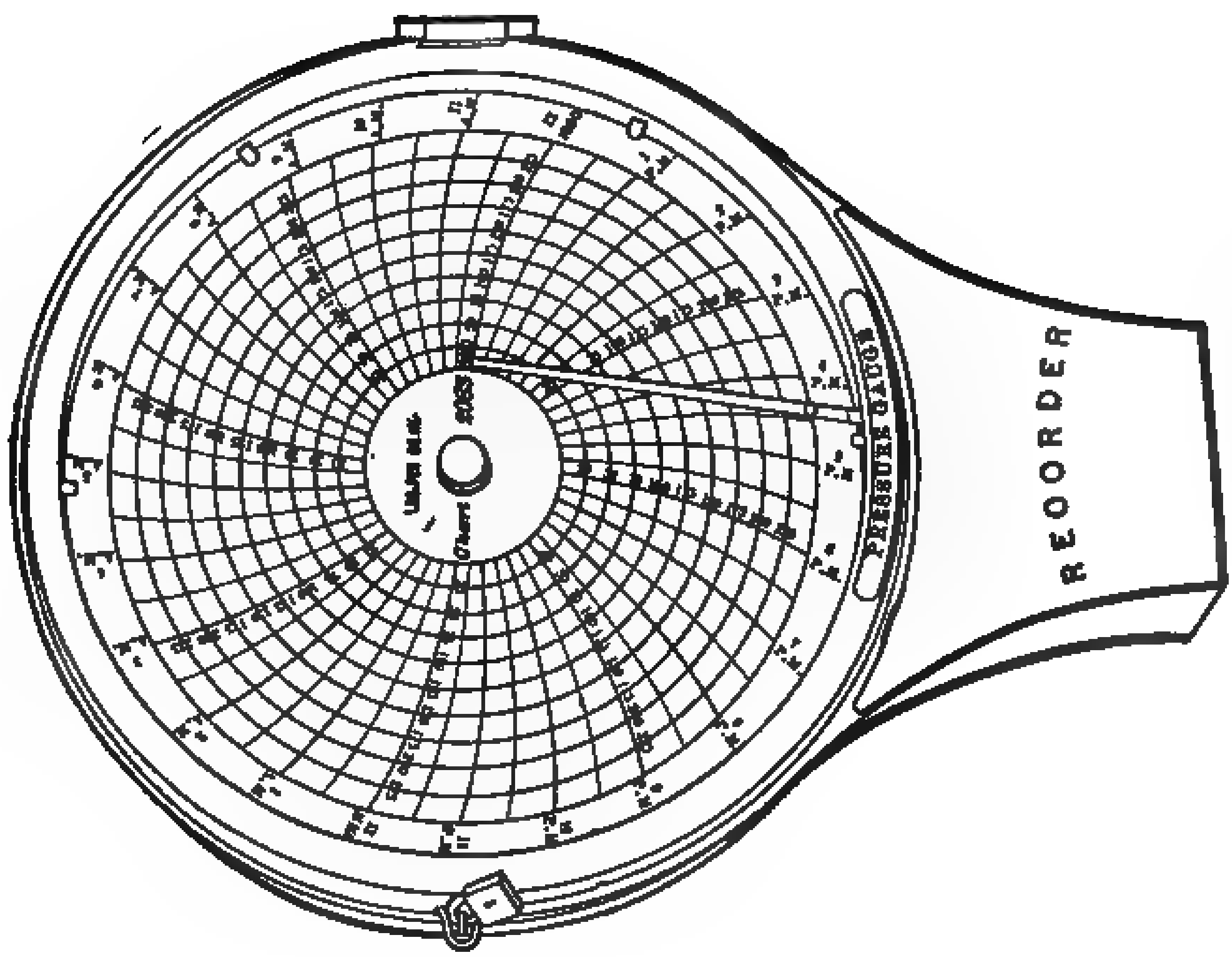


FIG. 20.—A Recording Pressure Gage.

FIG. 22. — A Recording Pressure Gage Operated by a Bourdon Tube.

FIG. 23. — A Low-pressure Recording Gage Operated by a "Diaphragm" Device.

These gages are made usually in one of the three following forms: (1) a circular tube of oval section in the form of a helix, as illustrated in **Fig. 21**, (2) with a metallic Bourdon tube, as shown in **Fig. 22**, or (3) with a diaphragm device, as in **Fig. 23**. In the instrument shown in **Fig. 23** the diaphragms spread out like an accordion when subjected to pressure on the inside. The first and second of these three types are generally used for cases where the maximum pressure is greater than three pounds and the third when it is less. The recording arm is preferably attached directly to the moving element so that no gears, levers, or other multiplying devices are needed. A more compact and less expensive form of such gages is illustrated in **Fig. 24**.

FIG. 24. — A Very Compact Type of Recording Gage. (Bristol.)

The **average pressure** corresponding to an irregular curve traced on the circular card of one of these recording gages is obtained with a fair degree of accuracy by integrating the curve by means of a Durand-Bristol integrating instrument described on page 87. Corrections to be applied to the readings of these gages are of course obtained by calibrating in the same way as for an indicating gage.

Still another type of recording pressure gage of the Bourdon tube pattern is shown in **Fig. 25**.

Calibration of Gages. Until recent years when the so-called "deadweight" apparatus for testing gages came into general use, gages used in other places than engineering laboratories were commonly calibrated by **comparison** with a so-called **test gage.** Such test gages have usually somewhat finer graduations than the ordinary gages used in practice and are probably also adjusted a little more accurately. They should never be exposed to the severe conditions of service, being intended only for purposes of comparison. This comparison of pressure gages[1] can be made anywhere by connecting the standard and the gage to be tested to any system of piping in which the pressure can be varied either

[1] Calibration of vacuum and low-pressure gages is discussed in another section, see pages 22.

FIG. 25. — A Combined Recording and Indicating Pressure Gage.

by pumping a liquid or by means of valves "throttling" steam, water or air under pressure. The only important precaution to observe is that the two gages shall be at approximately the same level when a liquid is used, and that the velocity of the fluid in the main pipe to which the gages are attached is negligible or is the same at the points where the connections for the gages are inserted in the "main" pipe. Test gages must, of course, be calibrated from time to time with some standard apparatus to insure their accuracy. A bench pump suitable for calibrating by comparison is illustrated in **Fig. 26.**

FIG. 26. — A Bench Test Pump.

Gage Testers. In very many power plants the use of the test gage has been superseded by some form of gage tester and by this means the gages used in the plant can be calibrated **directly** with an absolute standard. Calibrations of gages for high pressures by means of mercury columns are for practical reasons suitable only for laboratory work.

Dead-weight Gage Testers. The best-known form of this apparatus is made by the Crosby Steam Gage and Valve Co., and is illustrated in Figs. 27 and 28. The latter figure shows a partial section. It consists of a vertical cylinder **C**, into which is fitted very accurately a plunger **P**, of which the area, when new, is exactly one-fifth of a square inch. A circular platform upon which weights can be placed is attached to the upper end of this plunger. The cylinder **C** communicates at its lower end with the reservoir **R** fitted with an adjustable piston working in a screw and is operated by a hand wheel. A pipe **T** attached to the lower part of the reservoir is provided with unions and special fittings for attaching gages of various sizes. In the horizontal portion of this pipe there is a three-way cock or valve **V** for either draining the reservoir or for closing the pipe so that the liquid in the apparatus will not escape when the gage is removed. In operation, after the gage has been attached securely, the piston is screwed down to the bottom of the reservoir **R**, then with the plunger **P** removed, glycerine or heavy oil is poured into the cylinder **C** at the same time that the piston is screwed out. In this way the reservoir can be completely filled with oil without entrapping any considerable amount of air which would act as a cushion preventing the most satisfactory operation of the apparatus.

FIG. 27. — Crosby Dead-weight Gage Tester.

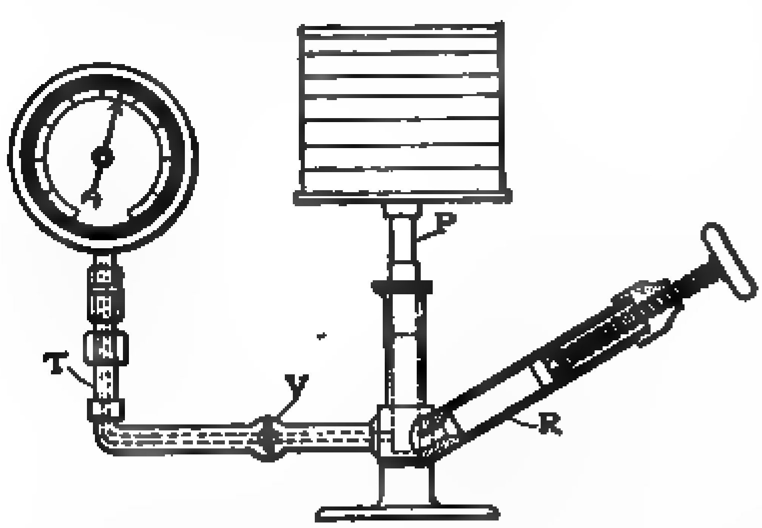


FIG. 28. — Section of a Dead-weight Gage Tester.

If the area of the plunger **P** is one-fifth of a square inch then each pound weight added on the platform produces a pressure on the liquid of 5 pounds per square inch. The weight of the platform and plunger (usually 1 pound) must always be included in the weight producing the pressure. As the load on the platform is increased the piston must, from time to time, be screwed in to keep the platform "floating." When observations are being taken it is very essential that the loaded platform be given, preferably by hand, a slight rotary motion to reduce to a minimum the friction of the plunger in its cylinder.

Suggested Procedure with Dead-weight Testers. The accuracy of the gage to be calibrated is determined by subjecting it to known pressures and noting its error. Before the plunger **P** has been put into place the reading of the gage, called "zero-reading," should be observed and recorded in a form similar to the one on page 21. Then the pressure should be increased 5 pounds per square inch at a time (corresponding usually to a **weight** of 1 pound) up to the limit of the graduations on the dial, **spinning the piston gently when each reading is taken.** Commencing then with the highest pressure the same operation should be repeated by decreasing the pressure by the same increments.[1]

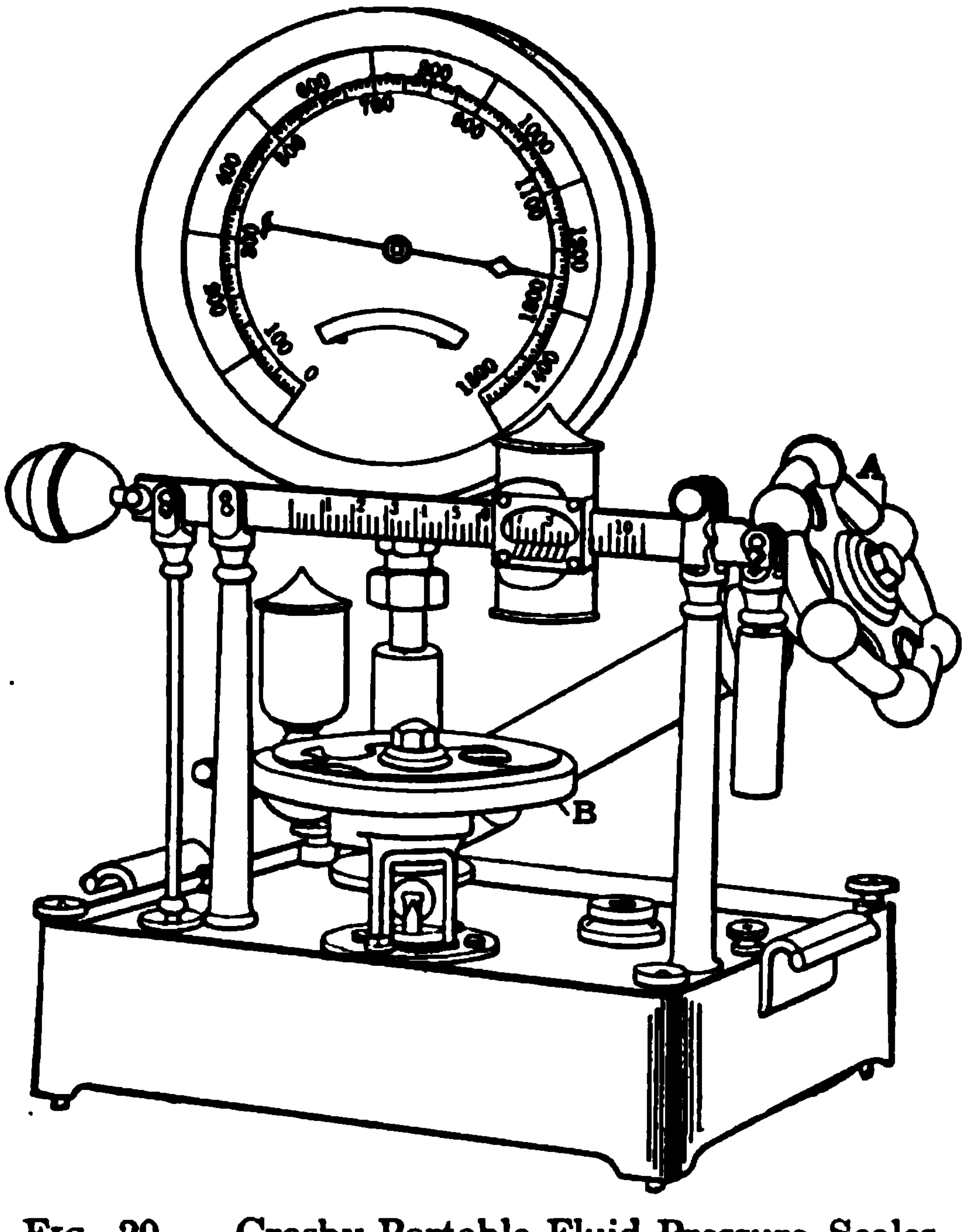


Fig. 29. — Crosby Portable Fluid Pressure Scales.

In case there is an appreciable difference between the area of the plunger **P** and its cylinder **C**, the two areas should be averaged to obtain the true area to be used in

[1] When the pressure is being decreased the movement of the pointer must be always in a counter-clockwise direction just before a reading is taken. In other words, if a weight of 2 pounds has been taken from the load on the plunger when only 1 pound should have been removed, the pointer will, of course, get below the next point to be calibrated. To secure the reading missed it will not be correct to add 1 pound and take the reading, because the friction and lost motion will now be in the same direction as with increasing pressures; and to overcome this difficulty the pressure must be increased again to a value higher than that for which the reading is to be taken. For

calculating the unit pressure. In other words, the true pressure exerted on the fluid in the apparatus in pounds per square inch is the **total** weight in pounds divided by this average area in square inches. (Report of Power Test Committee of A.S.M.E., Nov., 1912, page 1695.)

A modification of the dead-weight gage tester is shown in **Fig. 29.** This instrument is particularly suited for calibrations at high pressures. Its range is from 0 to 1500 pounds per square inch. Any pressure within these limits can be obtained without shifting heavy weights. Readings are taken when the scale beam is balanced. The hand wheel A is used to regulate the fluid pressure by means of a piston as in the apparatus shown in **Figs. 27** and **28.** The other hand-wheel B shown in the figure must be kept rotating when observations are taken. The slight jarring of the parts due to its rotation serves to make the friction as small as possible. Particularly in the case of gages that have had severe usage so as to be badly worn, it is necessary to tap the back of the casing of the gage lightly before each reading is taken to be certain that the pointer is moving freely.

FIG. 30. — Fluid Pressure Scales for High Pressures.

For still higher pressures up to 12,000 pounds per square inch, a heavy stationary type, shown in **Fig. 30,** can be used.

Calibration of Gages with Mercury Columns. The ultimate standard for the determination of reasonably high pressures is the mercury column, but the apparatus required is so complicated and occupies so much space that this method is suitable only for use in laboratories where it will have the attention of skilled observers.

For the purpose of calibrating steam gages mercury columns have been fitted up in a variety of ways. The simplest of these is the method of connecting the gage to be tested by means of a short tube to a "**closed**"

the purpose of increasing the weight it is not necessary to put on more weights, as additional load in such cases can be put on by the pressure of the hand.

The same precautions apply with even greater force to calibrations made with test gages or with a mercury column. With either of these instruments discrepancies may occur with increasing or decreasing pressures. In fact the only certain way to get satisfactory results with these instruments is to keep the pointer of the gage or the mercury column, as the case may be, moving continually in the same direction.

mercury well into the top of which a long glass tube has been inserted. The pressure can then be increased either by displacing some of the mercury in the well by means of the plunger in the mercury pump, shown at the right-hand side of **Fig. 31**, and forcing it up into the glass tube, or else by pouring mercury into the tube from the top as must be done in the apparatus shown in **Fig. 32**. Zero pressure for comparison is to be taken on the column at the same level as the center of the gage. Beginning then with 5 pounds per square inch pressure on the gage observe the corresponding

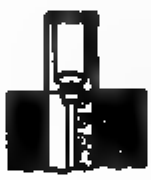


FIG. 31. — Standard Mercury Column and Hand Pump.

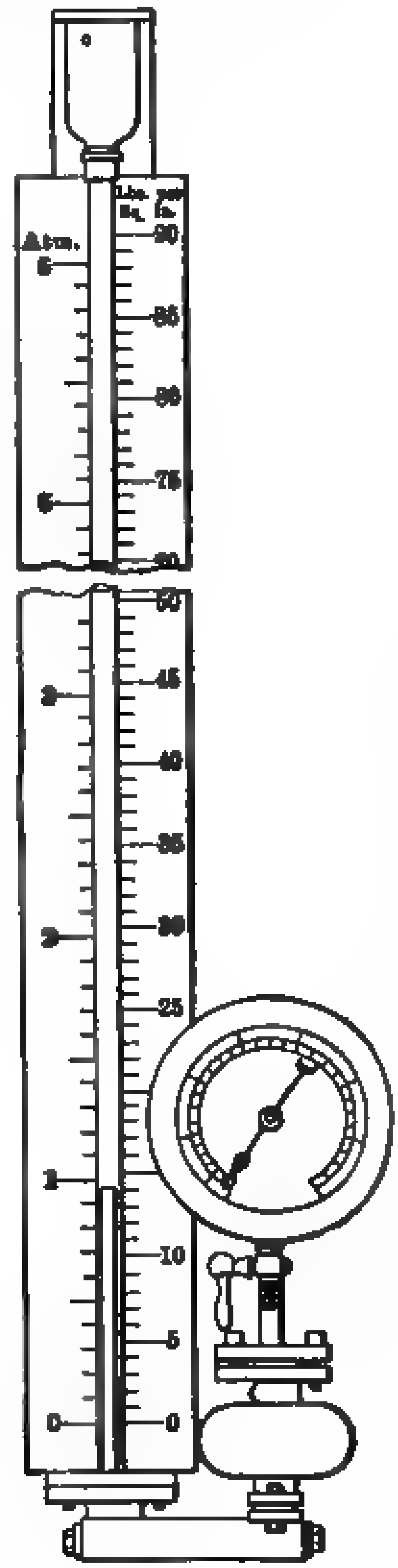


FIG. 32. — Simple Open Mercury Column.

height of the mercury column and its temperature, and then continue the observations, first increasing the pressure and then decreasing by increments of 5 pounds, as indicated by the gage. Equivalent units for calibration can be computed from the height of the mercury column,

since 1 inch of mercury at 70 degrees Fahrenheit is equivalent to a pressure of .4906 pound per square inch.

When using a mercury-testing apparatus it is necessary to observe the temperature near the mercury column in the room in which the work is being done, so that the observed height of the mercury column can be corrected to a temperature at which the relation between pressure in pounds per square inch and height is known. The coefficient of cubical expansion of mercury is not constant, as will be observed from the following table:

Temp. Deg. Fahr.	Coefficient of Cubical Expansion.
32	.0000998
50	.0001000
70	.0001002
90	.0001004
110	.0001007

For very accurate work allowance must be made for the linear expansion of the graduated scale. Coefficients of expansion of metals are given in the Appendix, Table II.

Instead of connecting the gage directly to the mercury well, it is sometimes connected to one end of a steam drum and the mercury column is connected to the other. The increments of pressure are then obtained by increasing or decreasing the steam pressure in the drum.

Observations taken in the calibration of a steam gage should be recorded and the computed errors tabulated in a form similar to the following:

CALIBRATION OF PRESSURE GAGE. COMPARISON WITH GAGE TESTER

Date............................Observers................................
Maker of gage.....................Maker's No..............................
Laboratory No....................Limits of Graduation......................

No. of Reading.	Weight on Tester. Lbs.	Gage Readings. Lbs. per sq. in.			Actual Pressure. Lbs. per sq. in.	Mean Error of Gage. Lbs. per sq. in.[1]	Remarks.
		Up.	Down.	Mean.			

[1] When the sign is +, the correction ("error") should be added, and when −, should be subtracted from the observed reading.

The error of the gage is determined by the comparison of the mean of the up and down readings with the **actual pressure.**

Curves. From the data tabulated two curves are usually plotted:

1. **Mean** gage readings (abscissas) and actual pressures (ordinates). Use a large sheet of coördinate paper for this curve. Unless plotted to a very large scale, however, such a curve will be of little value.

2. **Error Curve:** Mean gage readings (abscissas) and mean corrections, positive and negative (ordinates). See curve in **Fig. 33.** **Error curves** should have the points, if very irregular, connected by a broken line rather than by a "fair" or average curve through them. Never, however, try to draw an irregular curve through each of a number of scattered points when the points are supposed to follow a definite relation

Fig. 33. — Typical Error Curve for a Pressure Gage.

or law between the coördinates selected. A "fair" curve should then be drawn **between** the irregular points.

Calibration of Vacuum and Low-pressure Gages. A vacuum gage is usually calibrated by connecting it to one end of a U-shaped glass tube of which both legs are about 30 inches long and are filled to about half their length with mercury. The U-tube and gage are then connected to the receiver of an air-pump or else to an aspirator or ejector operated by water or steam pressure, such as chemists use for vacuum filtering. The aspirator (**Figs. 276** and **277,** page 236) is really the more convenient instrument to use. If the readings of the vacuum gage are correct, they will correspond exactly with the difference in the level of the mercury in the two legs of the U-tube.

In case a condensing engine is operating when the calibration of the vacuum gage is to be made, both the gage and the glass U-tube may be

connected to the condenser. A comparison of the readings taken will show, under the best possible conditions, the absolute errors of the gage. A suitable scale about 30 inches long and accurately graduated should, of course, be provided and placed between the two legs of the U-tube.

An apparatus consisting of an air-pump designed for a high vacuum and a mercury column is illustrated in Fig. 34. It is a very convenient means for testing vacuum gages.[1]

FIG. 34. — Air Pump and Mercury Columns for Testing Vacuum Gages.

A low-pressure gage with a scale from say 0 to 15 pounds per square inch is very easily and accurately calibrated by using the same glass U-tube mentioned for the calibration of the vacuum gage with air pressure,

[1] When measuring inches of vacuum by means of a single mercury column dipping into a cup, the zero is to be taken at the level of the mercury in the well into which the glass vacuum tube enters. The elevation of the gage above the surface of the mercury in the well is not to be considered, as the weight of the column of air between the center of the gage and the level of the mercury is negligible. Regarding the use of vacuum gages on water suction pipes see sections on Hydraulic Machinery.

preferably, or with steam pressure. Otherwise the method for calibration is the same as for a vacuum gage, except that inches of pressure instead of inches of vacuum are observed.

Draft Gages. Many engineers use an ordinary glass U-tube manometer (**Fig. 35**) filled with water for measuring small pressures like that due to the draft in a chimney or that produced in air-ducts by ventilating fans or blowers. For such observations in many cases, however, greater accuracy is desired than can be secured by the use of the ordinary U-tube and a special form of manometer is used in which the distance moved by the surface of the liquid in the tube is greater than the vertical change of level. **Fig. 36** illustrates a simple device of this kind. It consists of a bottle **B**, filled with water, having a suitable opening at the bottom to which by means of a short rubber tube the inclined glass tube **CD** is attached. At the upper end of this tube a piece of rubber tubing **T** is shown and is intended to be connected to the chimney, duct, or flue in which the pressure is to be obtained. A scale placed behind the inclined tube **CD** should be graduated so that when the spirit level **L** is adjusted, **vertical** differences in level of the liquid in **CD** will be indicated by its graduations. Then differences in the readings of the scale will give directly the difference in pressure in inches of water just as with an ordinary U-tube. Unless the bottle **B** is very large in diameter it is best to mark the graduations on the scale for every half inch by comparison with a U-tube[1] like **Fig. 35**. Intermediate divisions can then be marked with the help of dividers.

FIG. 35. — Simple U-tube Draft Gage.

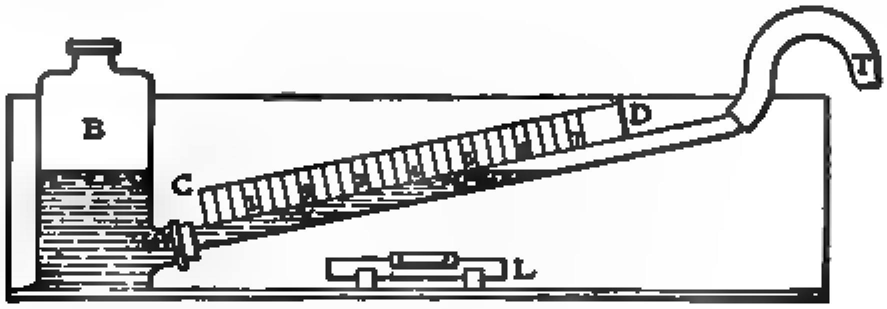


FIG. 36. — Inclined Tube Draft Gage.

[1] This method for graduating and also for checking the graduations is recommended by the Power Test Committee of A.S.M.E. For this special service the use of alcohol is advised instead of water in the U-tube, but allowance must be made for the difference in specific gravity between water and alcohol. It would be impracticable to use alcohol in gages that are in continuous service as the ordinary draft gage must be open to the air at one end, and the alcohol would very rapidly vaporize.

Very accurate draft gages of this type, known as Ellison's, are shown in Figs. 37 and 38. The inclination of the tube in these instruments is usually about 1 to 10. Instead of water a very light oil is used to fill the tube. It is claimed that this oil has the advantages of having less capillarity than water, and also, being lighter, permits the use of a longer scale

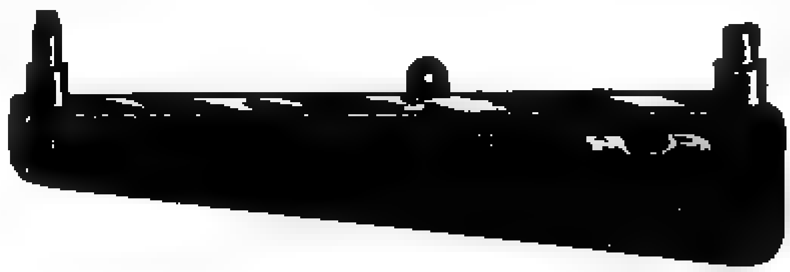


FIG. 37. — Ellison's Improved Draft Gage.

for a given difference in level. Graduations on these instruments which are sold commercially[1] are, however, always made to read equivalent inches of water. These draft gages are also often used for measuring small differences of pressure. For example, if there are two vessels containing gases at different pressures and one is connected to the left-hand side and the other to the right-hand side of the gage, it will indicate the difference in pressure. When used in this way it is called a differential gage (see page 12).

FIG. 38. — Ellison's Differential-direct Draft Gage for High Drafts.

Graduations on all such instruments should be checked by comparison with a simple U-tube, filled with distilled water (condensed steam from a surface condenser is satisfactory). The U-tube should be provided preferably with a steel scale of a recognized standard make.

When calibrating gages it is worth while to notice that when instruments are to be used to observe practically constant values it is necessary to calibrate them only near the values to be observed.

[1] American Steam Gage and Valve Mfg. Co., Boston and Chicago.

Draft gages which have liquids of slightly different densities combined so as to magnify the difference in level are sometimes used. Generally such an instrument consists of a U-tube with the top ends of very large area[1] and the connecting portion of small capillary tubing. Obviously the liquids used must have very little tendency to mix together or to become diffused. **Fig. 39** shows one type of such gages. As illustrated the U-tube was first partly filled with the heavier liquid, and afterward a lighter liquid was poured into each leg so as to completely fill the capillary and also fill **partly** each of the vessels **A** and **B**. The level of the separating meniscus on either side can be used as the indicator. Since the area of the capillary is very small compared with that of the vessels **A** and **B**, the indicating meniscus receives quite a large displacement for small differences in pressure. If the liquids used are kerosene (sp. grav. = .80) and brandy (sp. grav. = .94) the displacement of the separating meniscus due to pressure or vacuum will be about seven times the equivalent inches of water. Instruments of this type are usually graduated to indicate the pressure in inches of water.

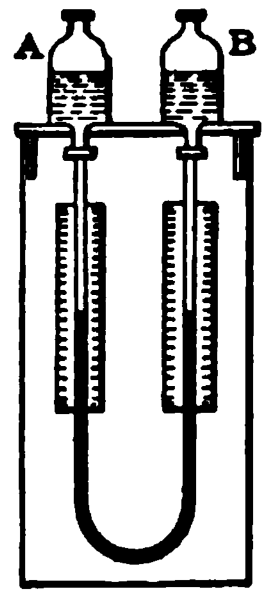


FIG. 39. — Simplest Two-fluid Type of Manometer.

A very delicate draft gage can be made also by using the apparatus of **Fig. 39** in a little different way, as in **Fig. 40.** **Water** is put into **A** and oil into **B**, with the surface of separation originally at **Q**, when **A** and **B** are both subjected to the same pressure. When **B** is connected to a chamber in which there is a vacuum as in the breeching or the chimney of a power plant the levels are changed from **F** to **E** in **B** and from **C** to **D** in **A**. The surface of separation moves from **Q** to **R**. Let **CD** = **EF** = **x** inches, **QR** = **y** inches, and **s** = specific gravity of the oil, then the change of pressure **p** in inches of water in **A** and **B** is

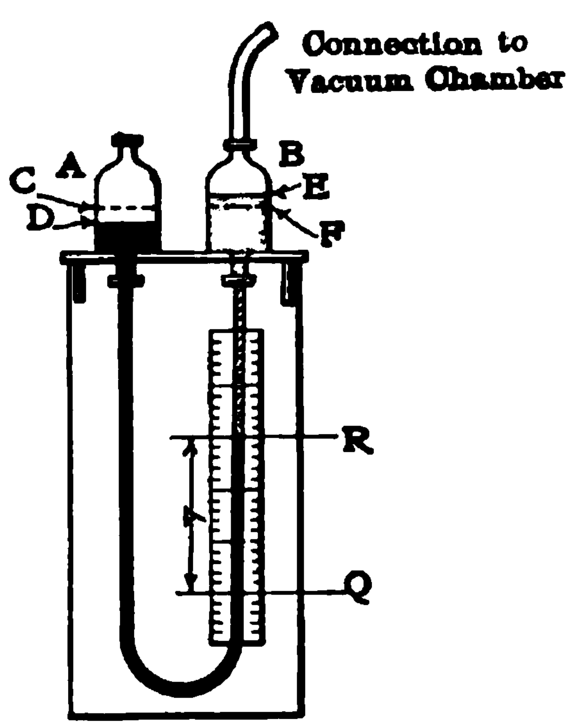


FIG. 40. — Another Type of Two-fluid Manometer.

$$p = x + sx + y(1 - s).$$

If the areas of **A** and **B** are equal and are **n** times the area of the connecting tube, then $y = nx$, and $p = \frac{y + sy}{n} + y(1 - s) = y\left[\frac{1 + s}{n} + 1 - s\right].$

Observe that the smaller the factor in the brackets the greater will be the magnification or sensitiveness. The value of **n** should therefore be made

[1] Preferably these top ends should be made with constant sectional area.

as large as possible, say about 100, and the two liquids should be preferably of nearly the same specific gravity. Gasoline and brandy are frequently used for the two liquids. They are of nearly the same specific gravity and the line of separation is readily observable. If S is the specific gravity of the brandy and s that of the gasoline, obviously

$$p = y\left[\frac{S + s}{n} + S - s\right].$$

Values of pressure causing a given movement of the line of separation may be calculated from the equations above, but actual calibration by comparison with an ordinary U-tube and graduation by exterpolation as recommended by the American Society of Mechanical Engineers is always preferable.

These last types of draft gages are too delicate for ordinary service in power plants to measure draft, but are particularly well suited for use with Pitot tubes in measuring air velocities as discussed in Chapter VII.

CHAPTER II

MEASUREMENT OF TEMPERATURE

Mercurial Thermometers. Temperatures less than about 500 degrees Fahrenheit are usually measured by means of mercurial thermometers, depending for their action on the expansion of mercury in a glass bulb and a graduated capillary tube.[1] The ordinary type is made so as to have merely a vacuum in the capillary tube above the mercury. For use at higher temperatures the capillary may be filled with nitrogen gas when the thermometer is made. It will then be serviceable up to 1000 degrees Fahrenheit. Thermometers made of quartz instead of glass and filled with nitrogen or carbonic acid gas can be used for temperatures as high as 1500 degrees Fahrenheit. Quartz thermometers are much stronger than those made of glass but are too expensive for ordinary commercial use.

Whenever mercurial thermometers are used for any work where reasonable accuracy is expected they should be carefully calibrated before the test is made; and after the test the calibration should be at least roughly checked to be sure that the zero of the thermometer has not changed. Too often it happens in practice when tests are being made, as for example of a boiler or of a steam turbine, that in some way a thermometer not previously calibrated has been used, and before the end of the test is broken. It is then too late to get a calibration and sometimes very important results of tests are made doubtful because of such negligence. The "Instructions Regarding Tests" of the American Society of Mechanical Engineers require that all thermometers used shall be calibrated both before and after every test where important data are to be obtained.

Calibrations of thermometers of all kinds must be made often because there is always the possibility that a little of the mercury has become detached from the column and remains unobserved either on the sides

[1] The best thermometers for ordinary engineering work are those having graduations etched on the stem and filled with ink. This is the only type of thermometer recommended by the Power Test Committee of the A.S.M.E. After considerable use, however, the ink originally in the etched markings disappears and it becomes difficult to read the scale. When this happens a thick paint made of lampblack and shellac or printer's ink can be rubbed over the etched scale, and when this paint is rubbed off, after a few minutes, there will be enough of it left in the etchings to make the scale as legible as when new. A crayon or pencil of soft greasy graphite like those used by glaziers for marking glass or by shippers for marking cases is a satisfactory substitute for the paint, although it is not so permanent.

or at the top of the capillary tube. In all glass thermometers there is always taking place with use and time a gradual and permanent change in the volume of the bulb, more in new thermometers than in old ones, altering the zero point and, of course, also the true values for all the graduations.

Alcohol Thermometers. For the measurement of temperatures much below zero Fahrenheit thermometers filled with mercury are not satisfactory, and alcohol or "spirits of wine" is used. These liquids, on the other hand, are not suited on account of their high vapor tensions for high temperatures.

Conversion of Temperatures and Heat Units. Temperatures in Centigrade degrees are converted into Fahrenheit by multiplying by $\frac{9}{5}$ and adding 32. Kilogram-calories[1] multiplied by 3.968 give the equivalent British thermal units (B.t.u.), and kilogram-calories per kilogram $\times$ 1.8 give British thermal units per pound. A "small" or gram-calorie is one-thousandth as large as a kilogram-calorie.

Calibration of Thermometers. Tests to determine the accuracy of thermometers are made by subjecting them to known temperatures and noting the errors. This is done usually in one of two ways:

1. By comparison with a so-called "standard" thermometer known to be accurate.

2. By comparison with temperatures corresponding to steam pressures.

Since the second method is not applicable for temperatures below the boiling point of water, it is not often used below 212 degrees Fahrenheit. For "low-reading" thermometers, therefore, the first method is generally used.

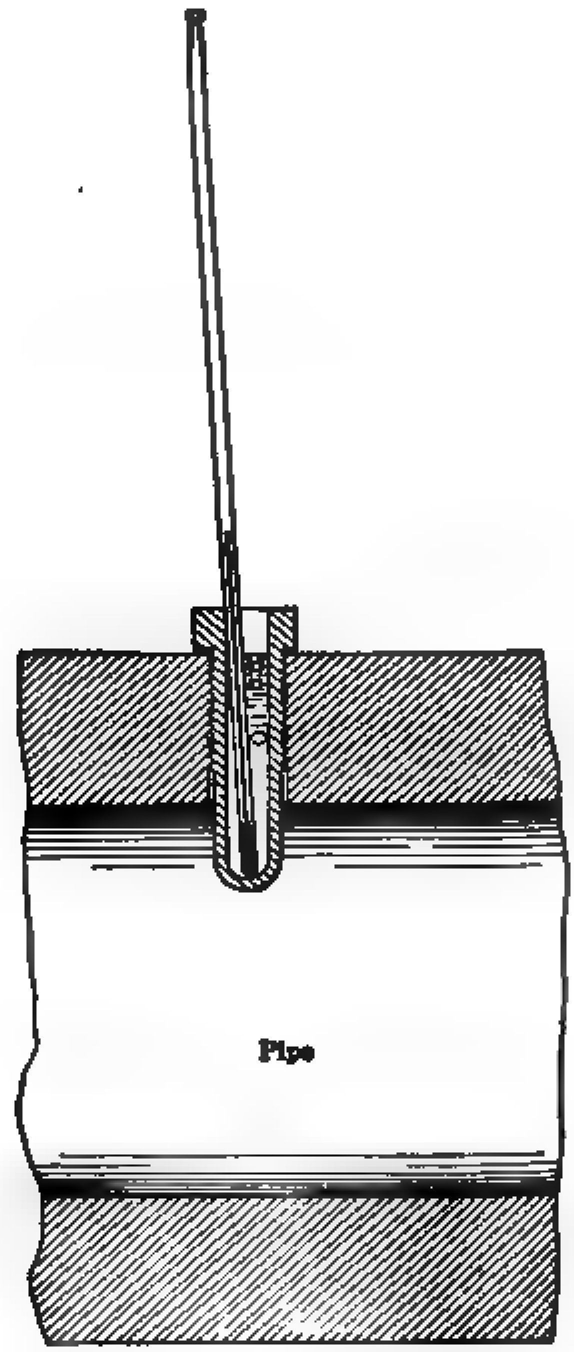
"Standard" thermometers for comparison should be preferably those which have been calibrated at standardizing laboratories, such as at the U. S. Bureau of Standards at Washington, D. C., at the Reichsanstat at Berlin, Germany, or at the National Physical Testing Laboratories in London, England. The steam laboratories of nearly all technical colleges have sets of standard thermometers suitable for determining the errors of good thermometers to be used as "secondary" standards.

After once calibrating a high-grade "standard" thermometer, it can usually be assumed that the calibrations throughout the range of its scale have not changed if the "ice-point" reading (for low-scale thermometers) and the "steam-point" reading at atmospheric pressure (for high-scale thermometers) remain unchanged. Checking either the "ice-point" or the "steam-point" at frequent intervals is the method ad-

[1] In German literature kilogram-calories are generally called Wärme Einheiten, usually abbreviated to W. E. For a scientific discussion of the distinction between "mean" calorie and "15 degree" calorie see Granberg's *Technische Messungern*, pages 250–252.

vised by the U. S. Bureau of Standards for determining whether any appreciable change has occurred in laboratory standards to make a new and complete calibration necessary.

The " ice-point " reading can be made by immersing the thermometer to the graduation marking 32 degrees Fahrenheit in good ice scraped from transparent cakes and mixed with the water obtained from its melting.[1]



The Right Way. The Wrong Way.

FIG. 41. — Typical Thermometer Wells — "Good and Bad."

Thermometers used at high temperatures should have the " steam-point " checked frequently in an apparatus like Fig. 43 (page 33).

The following paragraphs concerning standard thermometer wells, thermometers for high degrees of superheat, etc., are from the Nov.,

[1] See *Circular No. 8, Bureau of Standards.*

1912 Report of the Power Test Committee of the American Society of Mechanical Engineers:

"**Standard thermometers** are those which indicate 212 deg. Fahr. in steam escaping from boiling water at the normal barometric pressure of 29.92 in. (referred to 32 deg.), the whole stem up to the 212 deg. point being surrounded by the steam; and which indicate 32 deg. Fahr. in melting ice, the stem being likewise completely immersed to the 32 deg. point; and which are calibrated for points between and beyond these two reference marks.[1]

"A **thermometer well** consists of a hollow cup or plug threaded at the upper end and screwed into a threaded hole in the top of a horizontal pipe, the lower part extending vertically into the interior of the pipe as far, if practicable, as the center. The inside diameter should be slightly larger than the outside diameter of the thermometer tube and the well should be filled with mercury or high-grade mineral oil for temperatures below 500 deg. and with soft solder for higher temperatures.

"For superheated steam the portion of the well exposed to steam should be fluted or channeled so as to increase the area of the absorbing surface.

"Thermometers are so readily broken that it is desirable in important tests to have a sufficient number on hand that in case of accident the readings will not be interrupted. These spare thermometers should be calibrated in advance."

Experience has shown that certain types of thermometer wells for use in pipes give more satisfactory results than others. The thermometer well must be long enough to enter well into the pipe so that the flow of fluid through it will be around the well. In other words it should be located so that it will be in the " main stream " and not in such a position where only eddies touch it. A well-designed thermometer well is illustrated in the left-hand half of **Fig. 41**. On the right-hand side there is a correspondingly very poor design. To be sufficiently sensitive to temperature variations the bodies of such wells should be made of brass and the thickness of the metal where directly exposed to steam should not exceed $\frac{1}{16}$ inch. Steel and wrought-iron wells are, however, frequently used in ordinary commercial power plant work.

Calibration by Comparison with a Standard Thermometer. For low-temperature calibrations the thermometers to be tested are usually suspended together with a " standard " thermometer of which the errors are known in a water bath arranged so that the temperature can be varied. This bath may consist simply of a vessel provided with a coil of pipe through which steam can be circulated and has also a suitable stirring device. If the water is kept well stirred in such an apparatus a uniform temperature can be maintained and three or four thermometers can be calibrated at the same time.

[1] This definition of standard thermometers must also include any high-grade thermometers of which the errors throughout the scale are accurately known. It is practically impossible to make thermometers that will be absolutely accurate.

Fig. 42 illustrates diagrammatically a very simple apparatus of this kind, except that the water bath is heated by discharging steam directly into the water. This arrangement permits changing the temperature more rapidly than with the coil of pipe mentioned above.

When the method of comparison with a "standard" thermometer is to be used for temperatures higher than are obtainable with this apparatus, the "standard" thermometer and the other thermometers to be calibrated are placed in adjacent cups or wells inserted in a suitable cylindrical drum with pipe connections permitting a flow of steam around the thermometers. Fig. 43 shows a good design of this apparatus. Steam

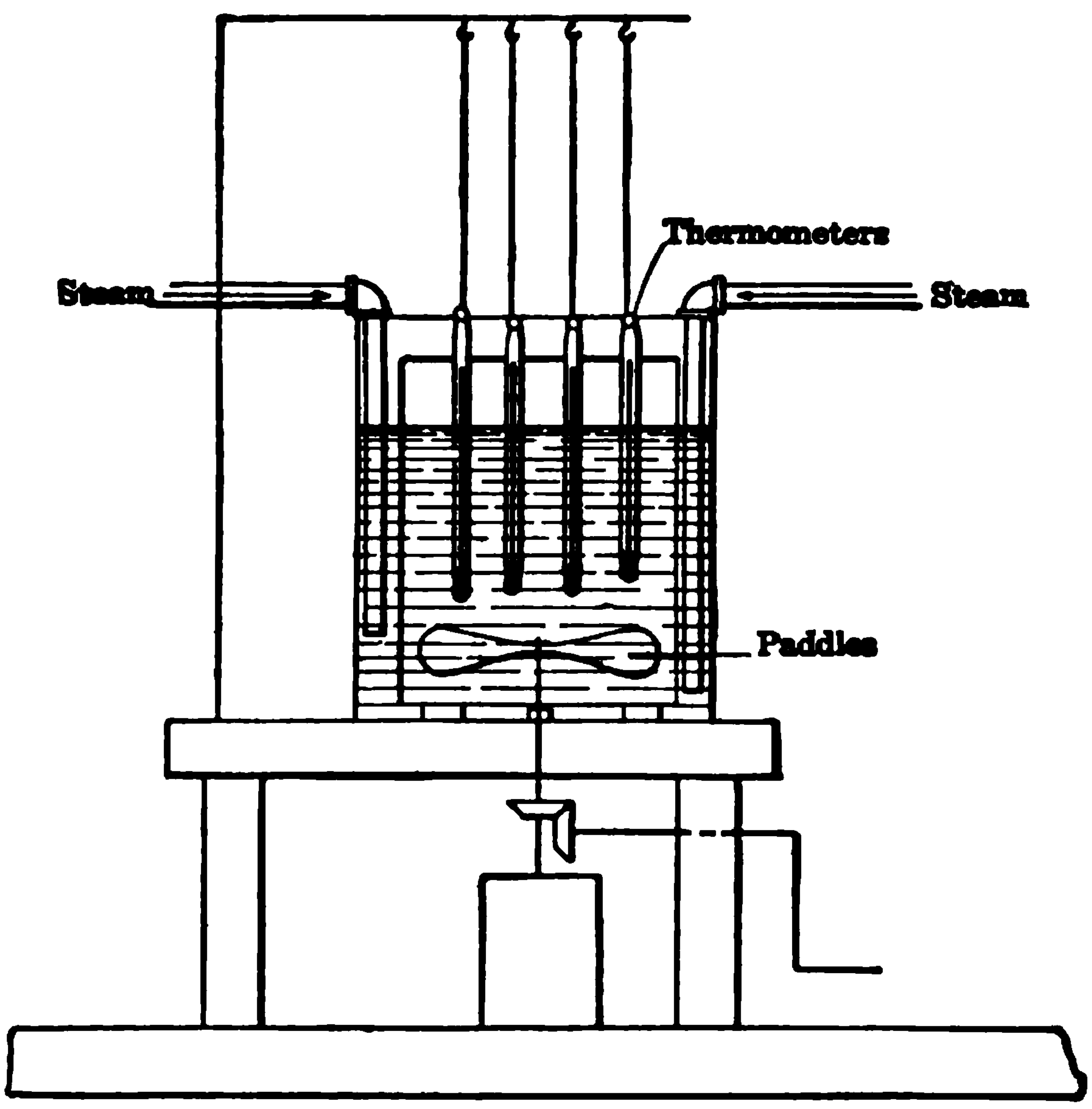


FIG. 42. — Apparatus for Calibration of Thermometers at Temperatures Less than the Boiling-point of Water.

passes through the regulating or needle valve A and enters the large cylindrical drum through the pipe C, which is open at the lower end. Exhaust is through D. A water gage is provided to show the level of water. Obviously if the bottom of the long thermometer well is immersed in water, the temperature indicated would not be comparable, as a rule, with the shorter thermometer exposed entirely to steam. The top of the exhaust pipe is made higher than the bottom of C so that there will always be some water "trapped" in the cylinder; and the steam which enters must bubble up through this water, making it more certain that the steam to which the thermometers are exposed is wet. In other words, the superheating which is likely to be caused by throt-

tling with the regulating valve A must be eliminated. Although only two thermometers are shown the apparatus can be very conveniently fitted with four or six thermometer wells so that a greater number can be calibrated at a time. The thermometer cups or wells should be filled with cylinder oil or, preferably, for high temperatures, with mercury.[1] Temperature is varied by throttling with the valves on either or both the steam inlet and discharge pipes. Usually the necessary adjustment is made more easily by manipulating the discharge valve rather than the inlet. At least five minutes should be allowed after the valves

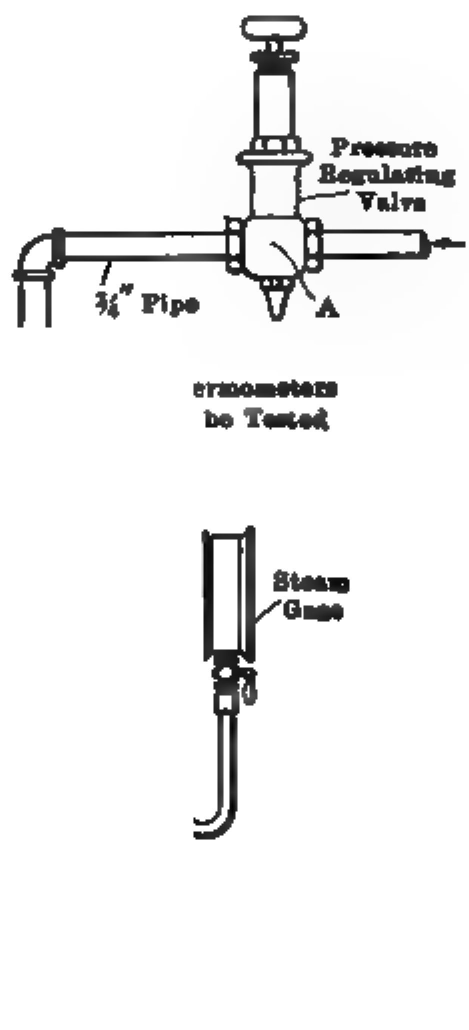


FIG. 43. — Apparatus for Calibration of Thermometers with Saturated Steam.

have been adjusted for the mercury in the thermometers to come to rest before readings for comparison are taken. Readings of both the standard and the thermometer being calibrated must be taken as nearly as possible at the same time, and the thermometers should be lifted from their cups, when necessary, just enough to bring the mercury into view. Observations should be taken as quickly as possible to avoid errors due to cooling and should be made with approximately the same increments.

[1] If oil is used in thermometer cups, precautions should be taken that the oil is absolutely free from the presence of water and that there is no water in the cups. If

The method of calibrating thermometers for temperatures between 212 and 400 degrees Fahrenheit by comparison with steam tables, as explained below, is more generally used than the method above.

A record of the observations should be made in the following form:

CALIBRATION OF THERMOMETER

By comparison with "STANDARD"..
Date..............................Observers..................................
Standard: Thermometer Tested:
No. or distinguishing mark...........No. or distinguishing mark..................
Range of scale......................Range of scale..........................

No.	Standard Thermometer.		True Temperature. °F.	Thermometer Tested.		Remarks.
	Reading. °F.	Known Error.[1] °F.		Reading. °F.	Error.[1] (+ or −) °F.	

[1] When the sign is +, the correction ("error") should be added, and when −, should be subtracted from the observed reading.

Calibration of Thermometers by Comparison with Temperatures Corresponding to Steam Pressures. There is a definite temperature of saturated steam corresponding to every pressure. If, then, the pressure is known, the temperature corresponding can be obtained from "**Tables of the Properties of Saturated Steam.**"[1]

Thermometers to be calibrated are placed in adjacent cups or wells of the same depth inserted in a cylindrical steam drum and filled with cylinder oil or mercury. Pipe connections must be provided for the attachment of an **accurately calibrated** steam gage, and valves are needed at the ends of the drum for regulating the flow of steam through it. The

water is allowed to accumulate there may be a slight explosion, sometimes strong enough to throw the thermometer out of the cup.

Thermometers used for measuring high temperatures should be protected from contact with the metal of the cups by wrapping the stems at the mouth of the cup with cotton waste or with rings of cork.

[1] Marks and Davis' *Steam Table and Diagrams* or Peabody's *Steam and Entropy Tables* (revised 1909) are recommended.

apparatus used is the same as that explained for calibration with a standard at high temperatures. **Fig. 43** shows the apparatus complete with a steam gage attached.

It is very important that the thermometer being calibrated should be immersed in the well to the same extent as it will be when in use. It is a very good arrangement to have a standard type of well for all thermometers of the same length, so that if a thermometer has been calibrated in one of these standard wells the calibration will be applicable in any one of these wells, provided the "room" temperature does not vary widely. Effect of variations of "room" temperatures can be readily calculated by the method explained on pages 36 to 38.

Calibration sheets for use with this method are made up like the one on page 34, except that pressure by the gage[1] is recorded instead of the temperature indicated by the "standard" thermometer.

If the steam supplied to the cylinder is superheated, then it is necessary to provide a water-jacket around the steam pipe large enough to make the steam at least dry saturated or preferably slightly wet. Another device often used to change superheated steam to the saturated condition is illustrated in **Fig. 44**. In principle it is the same as **Fig. 43**. In this apparatus the steam passes down through the vertical supply pipe **S**, closed at the lower end, and escapes from perforations near the bottom to bubble up through the water contained in the chamber **A** and is carried away in the pipe **D**. In this way the steam can be made to lose enough heat to the water to reduce the superheat. No valves or other devices that have a tendency to throttle the steam and consequently superheat it should be placed between the saturating drum and the steam cylinder in which the thermometers are to be calibrated.

FIG. 44. — Device to Reduce the Superheat in Steam.

A most excellent method for making calibrations of thermometers by means of a steam drum is a combination of the last two methods described. That is, the corrections are calculated from the temperature

[1] Since thermometers are calibrated usually only for increasing temperatures, the gage corrections to be applied should be those corresponding to increasing pressures. If only the average corrections of the gage are available for comparison, then the gage should be tapped lightly on the back of its casing when each observation is taken. Jarring has the effect of eliminating to some extent the lagging effects due to friction, which are presumably eliminated entirely when readings are taken with both increasing and decreasing pressures.

corresponding to the corrected gage pressure, and are at the same time **checked** by comparison with a " standard " thermometer.

In all experimental work in engineering such methods of checking results cannot be too highly commended. Checking not only assists in the elimination of errors of observation as well as in calculations, but for the engineer it is his key to success.[1]

Observations should be tabulated in the form given on page 37. The column under the heading " Standard Thermometer " is to be left blank, of course, when only the steam gage is used for the calibration. To obtain the **absolute pressure** the barometric pressure must be observed during the calibration.

Curves. 1. Plot a curve for each thermometer showing observed temperatures of the thermometer tested (abscissas) and the corresponding " true " or " standard " temperatures (ordinates). This curve is of no value, however, unless a reasonably large scale is used.

2. Plot an **error curve,** taking observed temperatures of thermometer tested for abscissas and " errors " for ordinates. Compare with **Fig. 33.**

Correction for " Stem Exposure " of Mercury Thermometers. When a considerable portion of the mercury column of a thermometer measuring high temperatures is exposed to the air, a correction **K** must be added to the readings to obtain the true temperature. If **t** is the observed reading, **D** is the number of degrees on the scale from the surface of the oil or mercury in the thermometer well to the end of the mercury column, and **t′** is the temperature of the air surrounding the thermometer stem, then for temperatures in Fahrenheit degrees,

$$\mathbf{K} = .000{,}088\,\mathbf{D}\,(\mathbf{t} - \mathbf{t'}). \quad \ldots \ldots \ldots \quad (2)$$

This equation, it will be observed, includes three factors: coefficient of expansion, length, and temperature difference. The coefficient of expansion given in the equation is the difference between the **volumetric** coefficient of expansion of mercury (.000,100)[2] and the linear coefficient (.000,012) of the kind of glass ordinarily used for thermometers.

Recent work by Reimbach[2] shows that this equation is not quite accurate, particularly for comparatively short stem exposures. His determinations are given for Jena glass thermometers with degree graduations about $\frac{1}{20}$ inch long in **Figs. 45** and **46. Fig. 45** is for thermometers with the ordinary type of solid stem, that is, with the scale on the stem. **Fig. 46** is for " sleeve " thermometers with a capillary tube enclosed in an outer glass tube, which covers also the scale.

[1] The **plotting of curves** is still another means which the engineer uses continually for checking his calibrations, his tests, and his conclusions. It has been well said that "the physician buries his mistakes but that the mistakes of the engineer bury him."

[2] *Zeit. f. Inst.*, vol. 10 (1900).

CALIBRATION OF THERMOMETER BY COMPARISON WITH TEMPERATURES CORRESPONDING TO STEAM PRESSURES

Record:

1. Date and names of observers..
2. No. and type of gage..
3. No. of standard thermometer. . ...
4. Identification of thermometer tested..................................
5. Limits of graduation of both..
6. Barometer reading..................ins. mercury = ...lbs. per sq. in......

1	2	3	4	5	6	7	8	9	10	
		Observed Temperatures.		Pressures. Lbs. per sq. in.						
No. of Reading.	Time of Reading.	Standard Thermometer.	Thermometer Tested.	Gage Readings.	Barometric.	Corrected Gage Readings.	Absolute (6)+(7).	True Temp. from (8) and Steam Table °F.	Error of Thermometer Tested Compared with (9) °F.[1]	Remarks.

[1] When the sign is +, the correction ("error") should be added, and when −, should be subtracted from the observed reading.

Beyond the limits of the curves the equation on the preceding page should be used. One precaution should be noted to explain discrepancies. When equation (2) is used, t' is the temperature of the exposed stem, usually obtained by tying a very short thermometer to the stem of the one being calibrated. Reimbach's data were obtained by observing the temperature corresponding to t' with an "auxiliary" thermometer, having its bulb about four inches away from and on a level with the mid-point of the exposed stem. In the use of the equation and the curves for stem corrections the difference in the methods of observing t' must not be overlooked.

Example. The observed thermometer reading is 500 degrees, temperature of the stem is 70 degrees, and immersion is up to 300 degrees. All temperatures Fahrenheit.

Stem correction by formula (2) is

$$K = .000088 \times 200\ (500 - 70) = 7.57 \text{ degrees.}$$

Corrected reading is 500 + 7.57 or nearly 507.6 degrees Fahrenheit.

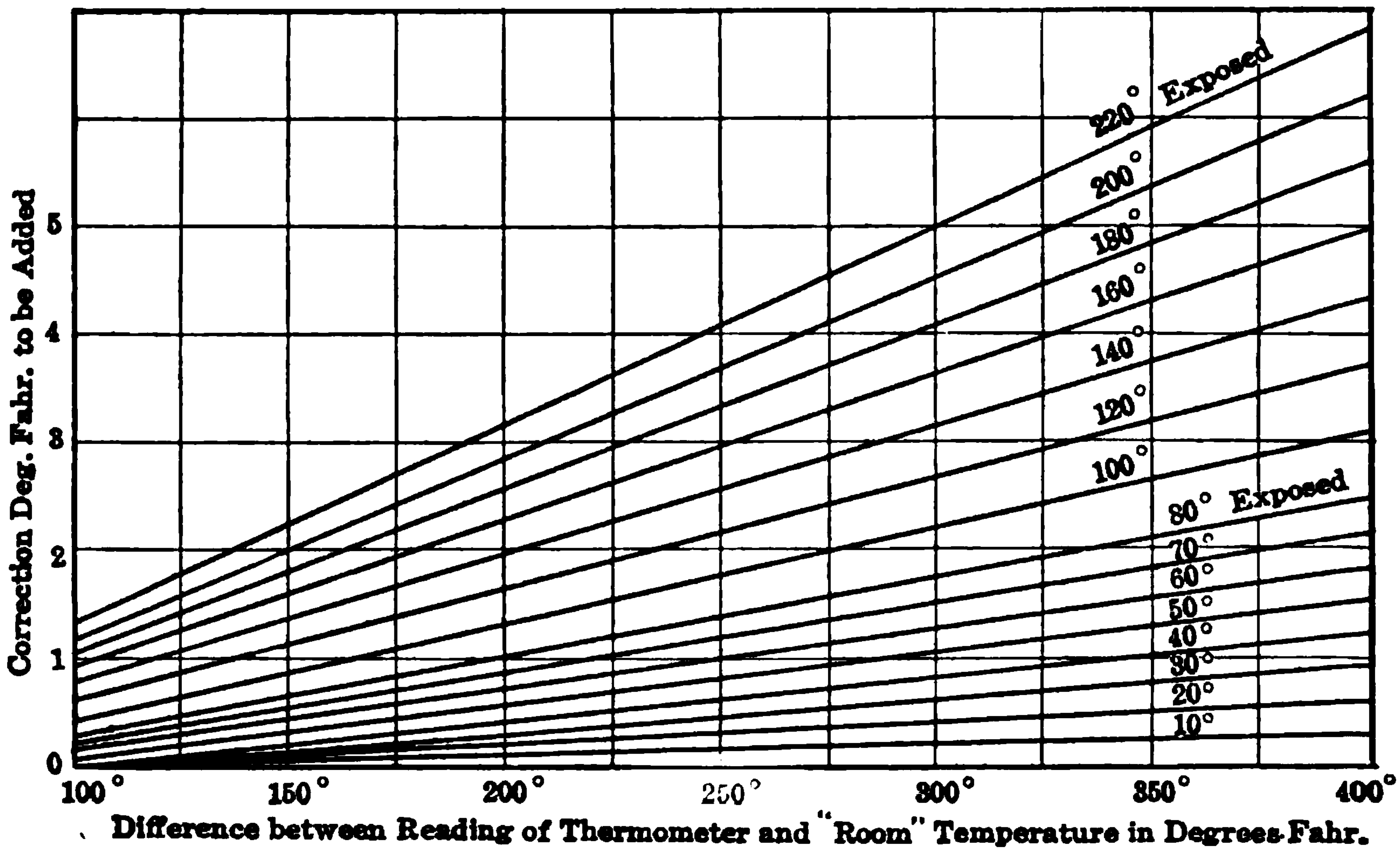


FIG. 45. — Exposure Corrections for Thermometers with a Solid Stem.

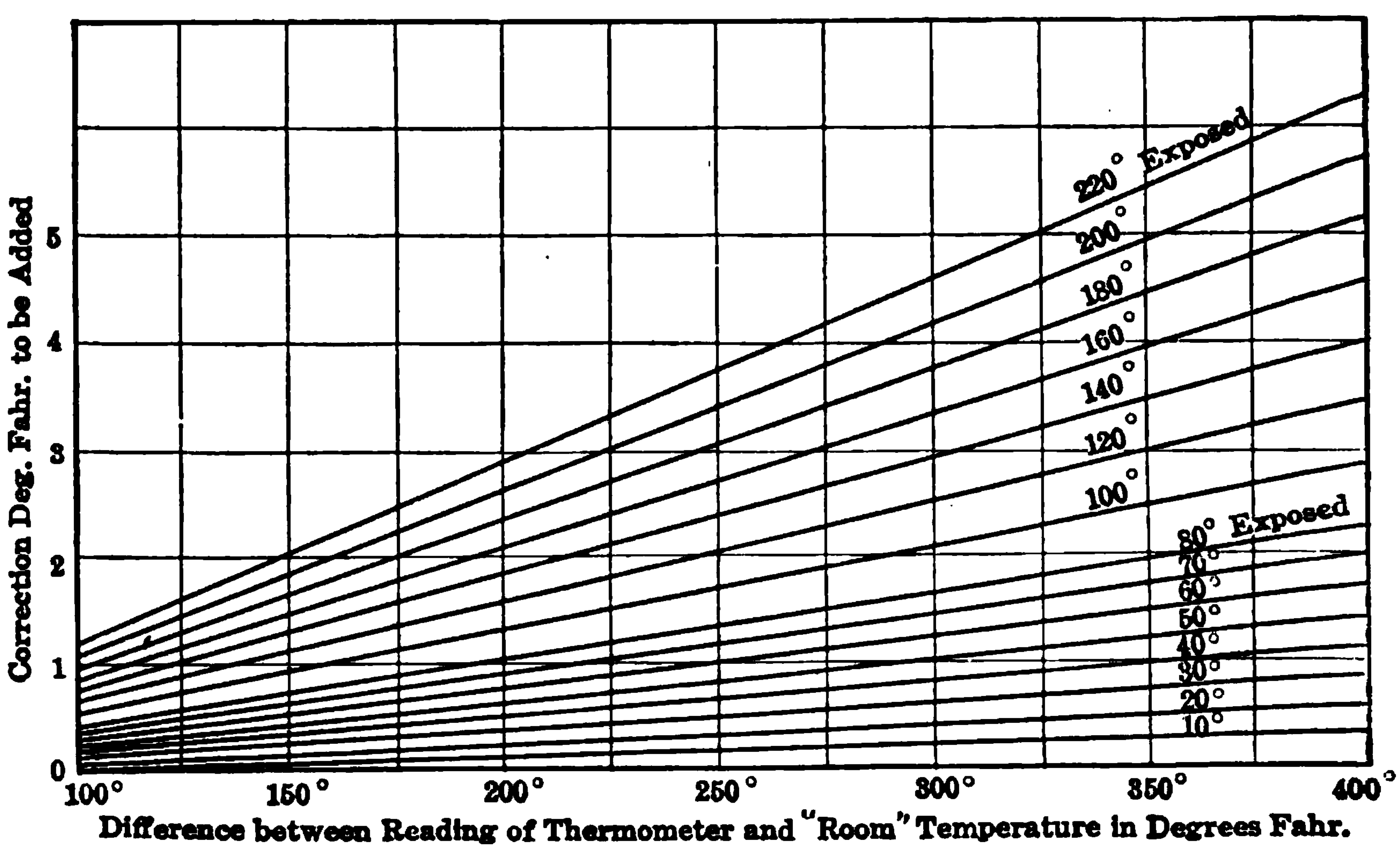


FIG. 46. — Exposure Corrections for Thermometers of the "Sleeve" Type.

In practice for carefully conducted tests of engines or turbines operating with superheated steam corrections as indicated should always be added to the thermometer readings to obtain the correct temperature

and superheat. In steam turbine tests, when a high degree of superheat is used, this correction is often as much as from 5 to 10 degrees Fahrenheit.

Recording Thermometers. Recently instruments for recording automatically low as well as high temperatures have been very satisfactorily developed. A typical example is shown in **Fig. 47.** It consists of a

FIG. 47. — Typical Recording Thermometer with Flexible Tube.

sensitive bulb (**Fig. 48**) suitable for being inserted into a pipe fitting and is attached by a capillary connecting tube to the recording instrument. The sensitive bulb and capillary tube are filled with either mercury or ether, which is sealed in the bulb and tube under pressure. The instrument is operated by the expansion of the vapor of these liquids.

Vapor thermometers consist essentially of a metal bulb partly filled with a liquid which, when heated, gives off a vapor which exerts a pressure

on a pressure gage through a small capillary tube. The following liquids are used, depending on the range of temperature:

Liquid sulphur dioxide (SO_2)	15 to 200 degrees Fahrenheit.
Ether (free of water)	95 to 250 degrees Fahrenheit.
Water	212 to 450 degrees Fahrenheit.
Heavy hydrocarbons	410 to 700 degrees Fahrenheit.
Mercury	650 to 1350 degrees Fahrenheit.

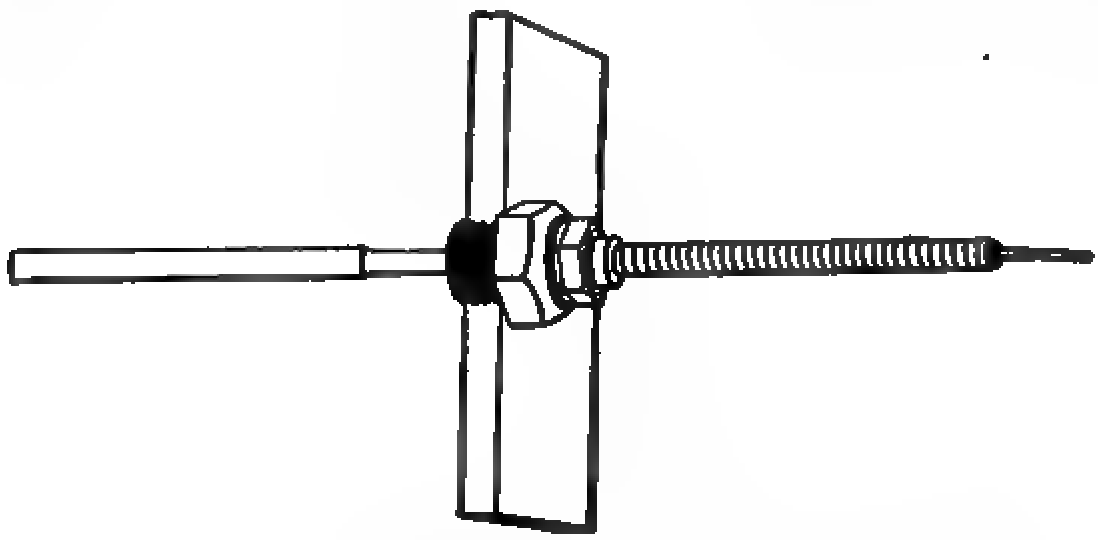


FIG. 48. — "Sensitive" Bulb for a Recording Thermometer.

The capillary tube may be made 100 feet long, and such instruments are suitable for "distant reading," but varying the temperature of the capillary tube by exposure will alter the observations. The whole length of the bulb must be exposed to the temperature to be measured, and complete immersion of the bulb is sometimes difficult in lines of piping of small size. In instruments using mercury vapor the bulb has a volume of about one cubic inch in outside dimension per 100 degrees Fahrenheit range of temperature. Those filled with ether are of about half the volume required for mercury.

One of these instruments is shown in Fig. 49 with the cover removed so that the mechanism can be seen. It is exactly the same as that of a recording pressure gage (see page 13).

FIG. 49. — Mechanism of a Recording Thermometer.

In general appearance and in the operation of the recording mechanism and clockwork, these recording thermometers are like the recording pressure gages now in general use. Some of these recording instruments, Fig. 50, have a short rigid connection between the bulb and the recording mechanism, making it necessary to locate the instrument always immediately adjacent to the bulb. In Fig. 47 there is a flexible connection of capillary tubing attached to bulb permitting the setting up of the

FIG. 50. — Recording Thermometer, Short Bulb Type.

instrument on a wall near by. This capillary tube must, however, be handled very carefully to prevent causing a serious leak, making the instrument useless.

Pyrometers. Temperatures over 600 degrees Fahrenheit are usually measured by instruments known as pyrometers. Various types are in use particularly for the measurement of temperatures in flues and chimneys of boiler plants.

Thermo-electric Pyrometers. When two wires of different metals are joined at both ends, so as to form a complete metallic circuit, as in Fig. 51, and if the two junctions H (hot) and C (cold) are at different

temperatures, an electro-motive force is generated which can be measured with a galvanometer or commercial milli-voltmeter. If the cold junction is always maintained at a constant temperature the scale of the galvanometer can be graduated to read directly the temperature of the hot junction. In practice it is usually impracticable to maintain a constant temperature at the cold junction so that usually a compensating device is arranged to eliminate the error. One of these devices (**Fig. 52**) consists of an air-tight glass bulb partly filled with mercury into the top of which a U-shaped platinum loop is fused. This platinum loop is long enough to extend into the mercury and its ends are connected to be in series with the thermo-couple at the cold junction. When the temperature of the leads or outside circuit falls, the voltage due to the couple increases because of the greater range of temperature at the "hot junction," but the mercury in the bulb contracts so that the current must pass through a greater length of the high-resistance platinum wire in the loop. The net effect is that the increased resistance neutralizes the greater voltage produced at the "hot junction." Another method of compensation is to attach a small mercury thermometer to the cold couple and put a wire resistance in series with the circuit, which can be cut out in varying amounts by adjusting a lever on a dial graduated to make the resistance correspond to the temperature.

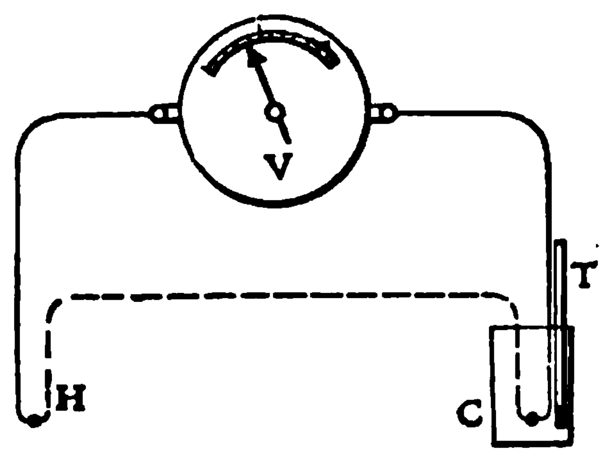


FIG. 51. — Diagram Illustrating Action of a Thermo-couple.

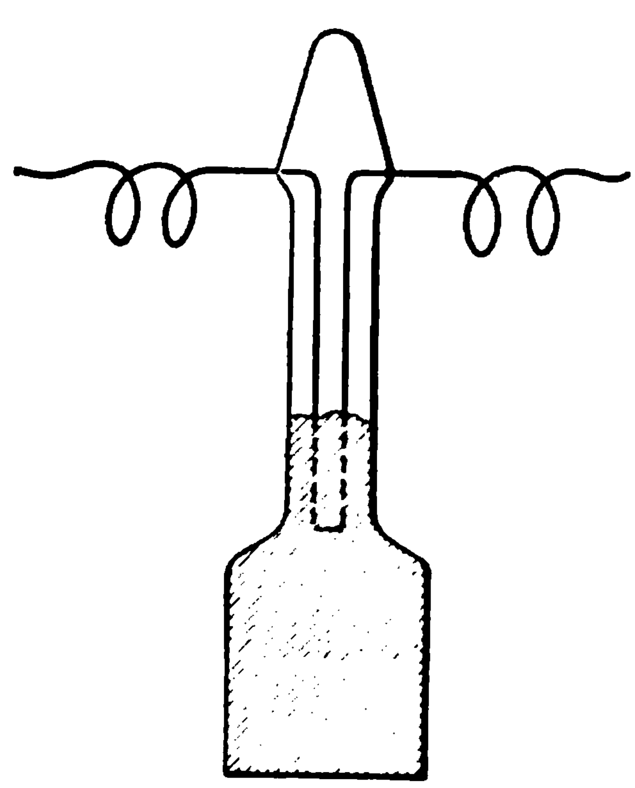


FIG. 52. — Mercury Compensating Device for Thermo-couples.

There are two general types of such instruments (1) high resistance; (2) low resistance. The high-resistance type has a couple formed of two wires of small diameter. One wire is of platinum and the other is an alloy of 90 per cent platinum and 10 per cent rhodium.[1] To protect the fine and delicate wires against breakage and also because platinum deteriorates in the silicon, phosphorus and other "gases of reaction," the couples of the high-resistance type are always protected by porcelain or iron tubes. If the temperature never much exceeds 1500 degrees Fahrenheit iron tubes are satisfactory. For higher temperatures porce-

[1] Formerly iridium was used for alloying but it volatilizes rapidly at over 1500 deg. Fahr., causing a gradual lowering of the voltage produced.

lain tubes are used, but they must be handled carefully as they are easily cracked. Base-metal or low-resistance types of couples are usually made of alloys of nickel, iron, and copper. Couples used very largely in America for temperatures up to about 1500 degrees Fahrenheit are made of nickel-steel and copper; another is made of one wire of nickel and the other of an alloy of nickel and chromium. Such couples made of cheap metals can be made of larger rods and at less cost than the high-resistance type. Even though they are not quite as accurate they are generally preferred for industrial work. There is also the advantage that a broken couple can be readily replaced by rods of **iron and copper** fused together at one end. A calibration curve for the couple is easily made to be accurate enough for practical purposes. The principal consideration in selecting rods for such couples is to get them of uniform chemical composition and they should be annealed preferably in an electric furnace to a temperature higher than that to which they are to be exposed. If rods in a couple are not of uniform composition, parasitic currents are produced which oppose that produced by the couple at the junction. Since these wires can be made comparatively large, usually about $\frac{3}{32}$-inch diameter, the current generated will be large compared with the high-resistance types and its change in resistance with change in temperature will be small, so that a cheaper low-resistance galvanometer can be used. Low-resistance couples are usually protected by an iron tube, mainly because the steel wires deteriorate in the presence of sulphur gases and the asbestos insulation needed for separating the rods along their full lengths is likely to last longer with this protection.

Low-resistance pyrometers have often the leads of the same metals as the couples so that the so-called "cold-junction" is at the terminals of the galvanometer, and the leads are then usually made long enough to permit the instrument being placed where the temperature can be maintained at about normal "room" temperatures. Variation in the "cold junction" temperature from the calibration temperature produces more error in low-resistance than in high-resistance types.

When iron is a constituent of a couple, it should not be used for temperatures above 1300 to 1400 degrees Fahrenheit as this temperature is a "transition point" for this metal and its physical and also its thermoelectric properties are changed.

Thermo-couples made of platinum and a platinum-nickel alloy produce twice the voltage of a platinum-rhodium combination, but it should not be subjected to temperatures above 2000 degrees Fahrenheit. If the couple is made of one wire of pure platinum and another of an alloy of platinum and about ten per cent of rhodium, temperatures nearly as high as the melting point of platinum, or nearly 3500 degrees Fahrenheit, can be measured, although 3000 degrees Fahrenheit is considered

the safe limit. This pyrometer with a platinum "couple" is known generally as a Le Chatelier type (**Fig.** 53).

Electrical Resistance Thermometers are based on the principle that the electrical resistance of some metals increases considerably as the temperature is raised. Platinum is usually selected because for a given temperature it has a remarkably constant resistance and it does not deteriorate at high temperatures. A resistance thermometer of the simplest type is made of a coil of pure annealed platinum wire **W** wound upon a mica framework (**Fig.** 54) in "series" with a very small coil in a casing **C** intended to be exposed to the temperature to be measured. The variation of resistance is measured by a Wheatstone's bridge method.

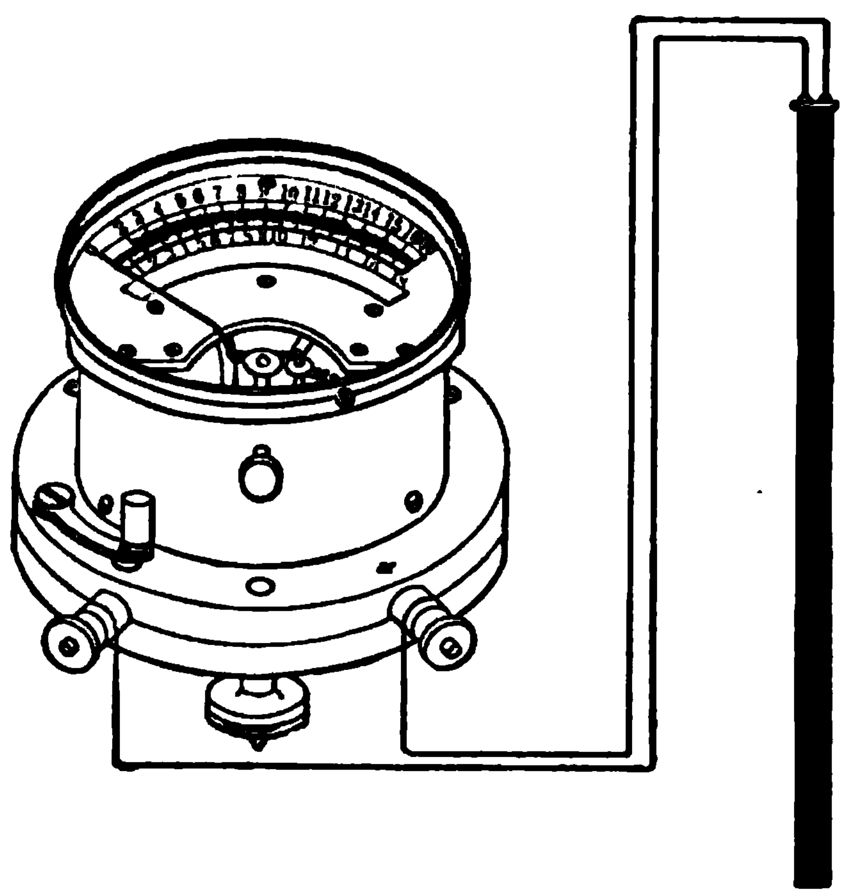


Fig. 53. — Le Chatelier Pyrometer.

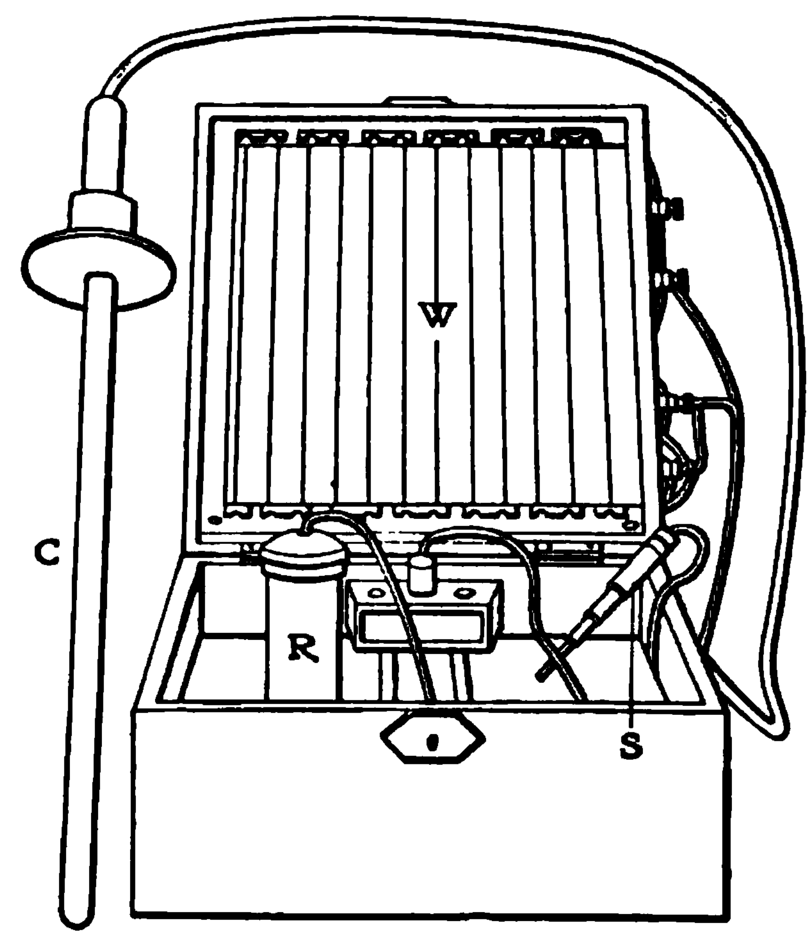


Fig. 54. — Resistance Thermometer Bridge.

The current from one electric battery passes through the wire in **C**, and the current from another battery passes through the coil **W** in the cover of the box. When the two circuits are connected so that the electromotive forces of the two batteries are opposed, the resistance in the cover is adjusted by means of a connection on a stylus **S** so that there is no current passing through a telephone receiver **R** or a sensitive galvanometer placed at the junction of the two circuits. For making observations this stylus is moved along the "scale" wire in the cover to a point where the humming noise due to the electric current ceases. The temperature can then be read on the graduated scale opposite the position of the stylus.

By means of a switchboard any number of "heating" elements can be connected to the same indicator box, which may be located at any distance from the source of heat.

Commercial instruments of this type are usually arranged so that the "bridge" indicates the temperature in degrees. Up to 1000 degrees Fahrenheit the error should be less than $\frac{1}{50}$ degree and at 2400 degrees Fahrenheit not more than $\frac{1}{2}$ degree. A delicate galvanometer sensitive for small currents is required. A large current in the necessarily small wires would by its own heating change the resistance and impair sensitiveness. For many classes of work, particularly if there is rough usage, the platinum coil C must be protected by a porcelain or iron tube. This protection introduces a time lag, so that very delicate instruments are not protected by a casing. The junctions of the platinum wire of the thermometer with the wires going to the resistance measuring device must be placed in the cooler part of the circuit, where the temperature should be the same as when the instrument was calibrated, or compensators may be used as explained on page 42. Electric resistance thermometers are readily calibrated at the temperatures of melting ice, steam at varying pressures (from 212 to 350 degrees Fahrenheit), and boiling sulphur (832.5 degrees Fahrenheit). Intermediate temperatures are computed.[1] Metals like copper, tin, and zinc when pure remain at a quite constant temperature for about a half hour when cooling slowly and passing from the liquid to the solid state.

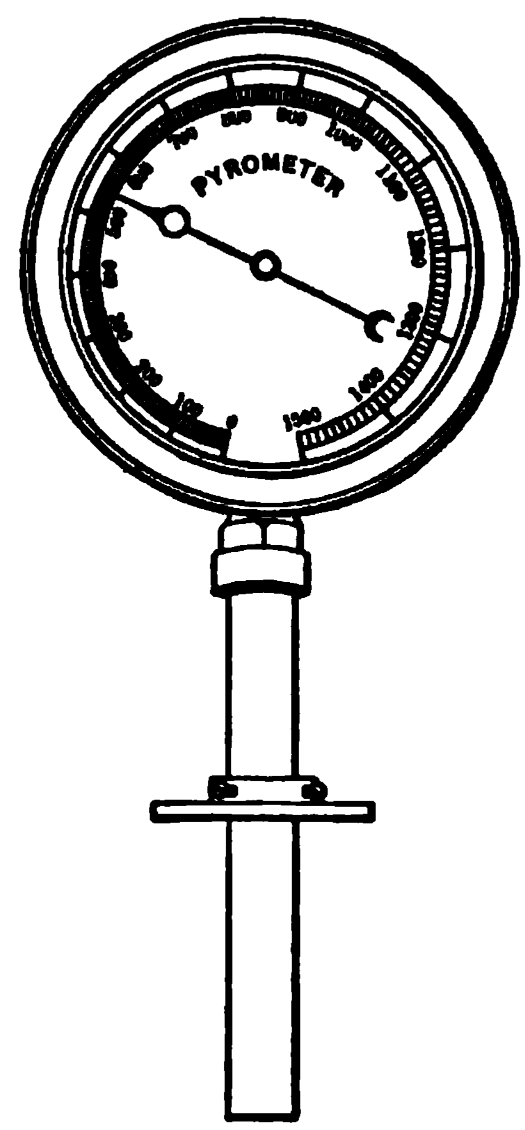


FIG. 55. — "Mechanical" Pyrometer.

Metallic or Mechanical Pyrometers (Fig. 55) consist essentially of two rods made of metals having different rates of expansion connected by gears and levers to rotate a pointer on a graduated dial. Generally the rods are made of iron and brass, or of graphite and iron. Although the use of such instruments is very common they are generally very unreliable, and should never be used for temperatures above 1000 degrees Fahrenheit. There is always a tendency for the zero of the instrument to get higher with use at even moderate temperatures. Beckert and Weinhold found that in a number of cases the zero changed from 200 to 400 degrees Fahrenheit in two months. In order to obtain readings corresponding to the graduations the entire length of the tube enclosing the rods should be placed in the chamber of which the temperature is being measured.

Calibrations of "Indicating" Pyrometers such as the **thermo-electric, resistance,** and **mechanical** types are best made by comparison with a

[1] Bulletin No. 7, U. S. Bureau of Standards.

special standard electric resistance thermometer of which the error is known and which is used only for standardizing work. The couple to be calibrated and the standard should be fastened together closely with only a sheet of asbestos between them. The two couples thus bound together should be put into an electric furnace in which the temperature can be controlled and raised very slowly. Then at different points in the scale, at intervals of about fifteen minutes, readings for comparison can be taken. If a standard resistance thermometer is not available a calibration can be made by comparison in a furnace of constant temperature with a good mercury thermometer.[1] Such thermometers in which the capillary tube contains rarefied nitrogen above the mercury can be obtained to measure temperatures with a fair degree of accuracy, when new, up to 1000 degrees Fahrenheit.

The method of calibration suggested by the Power Test Committee of the A.S.M.E. is as follows:

"Compare pyrometers for calibration at low ranges under proper conditions with a mercurial thermometer of known accuracy (both being placed for example in a current of hot air or flue gases of which the temperature is under control). Determine the errors at higher temperatures by plotting the results obtained as above on a chart, finding the curve of error, and continuing the curve to the higher ranges desired."

For extremely high temperatures such as that of a boiler furnace or the bed of coals in a gas producer, the radiation optical and pneumatic pyrometers may be used. (See pages 46 to 51.)

Pneumatic Pyrometers depend for their action on the variation of the flow of gases through orifices due to heating. Uehling's pneumatic pyrometer is shown diagrammatically in **Fig. 56.** As shown flue gas is continuously drawn through two orifices A and B by a constant suction produced by an aspirator **D.** So long as the air has the same temperature in passing through **A** as it has in passing through **B,** there is no change in the partial vacuum in the chamber between the two apertures; if, however, the air has a higher temperature when passing through **A** than when passing through **B,** the suction or vacuum in the chamber between the two orifices

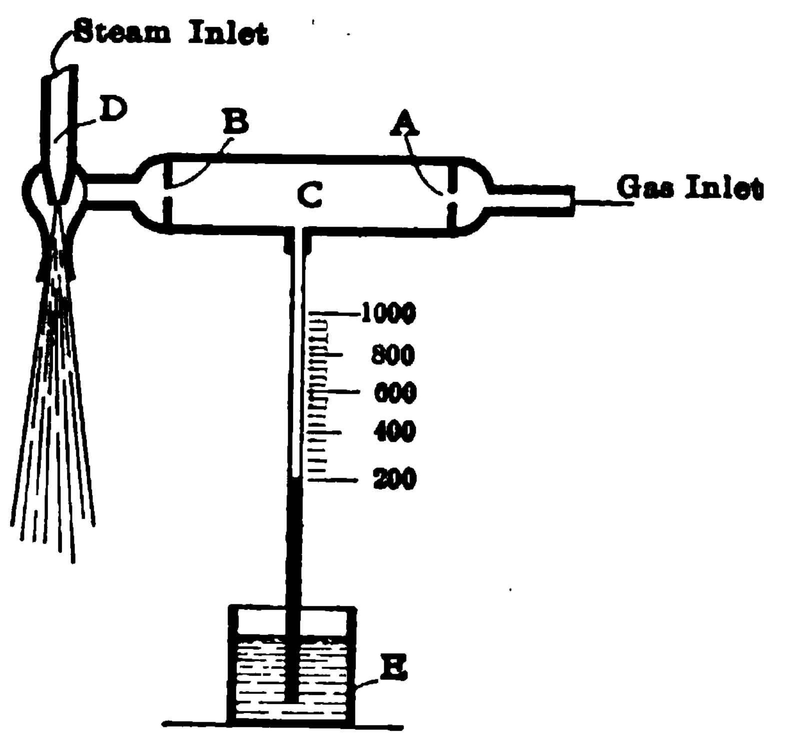


FIG. 56. — Pneumatic Pyrometer.

[1] Usually temperatures vary considerably inside a furnace so that the couple and thermometer should be bound together in order to be sure they are exposed to the the same temperature.

will increase in proportion to the difference in temperature between **A** and **B**, because the volume of air varies directly with the temperature.

In the application of this principle orifice **A** is located in a nickel tube which is exposed to the heat to be measured, while orifice **B** is kept at a uniformly lower temperature. Filters are provided for keeping the orifices clean. The instrument can be made to record and indicate the temperature at a distance. In order to maintain a constant vacuum or suction at B the steam pressure at the nozzle D must be maintained constant by means of a good reducing valve or other means.

Recording Pyrometers are most frequently of the type of recording thermometers illustrated and described on pages 39 to 41. Such instruments can be constructed, when the sensitive bulb is filled with a gas instead of a liquid, to register accurately temperatures as high as 1200 degrees Fahrenheit.

FIG. 57. — Combined Indicating and Recording Pyrometer.

Another type operated by the expansion of the vapor of mercury is shown in **Fig. 57**. This is a combined indicating and recording instrument. The sealed tube **A** is to be inserted in the chimney or flue in which the temperature is to be observed.

Radiation Pyrometers. For temperatures above 2500 degrees Fahrenheit radiation pyrometers similar to the one illustrated in **Figs. 58, 59** and **60** are most suitable. They can also be used in many places where it is almost impossible to locate a pyrometer of any of the other types.

The principle of operation is that the energy radiated by a so-called " black " body is proportional to the fourth power of its absolute temperature. The instrument illustrated consists of a cylindrical case set upon a tripod. This case contains a concave mirror and a lens (or lenses) which when properly adjusted and focused on a hot body concen-

FIG. 58. — An Optical (Radiation) Pyrometer in Use.

trate the heat rays upon a small thermo-electric couple[1] inside the case. Copper wires connect this couple with a very sensitive portable galvanometer (Fig. 59) located where it can be read conveniently. The most

[1] Some of these instruments have a metal coil made up of a pair of strips of metal of widely different coefficients of expansion which replaces the thermo-couple. The principle is the same as in the metallic pyrometers (page 45).

modern instruments of this kind are provided with scales indicating directly degrees of temperature. Fig. 60 shows a section of the telescope used in connection with this pyrometer. The concave mirror **M** receives the heat rays and focuses them at **F**, where a small thermo-couple is located. To assist in pointing the telescope an eye-piece **E** is provided through which a reflected image of the hot body can be seen. The rack **R** and the pinion **P**, moved by a thumbscrew outside the case, serve for adjusting the focus of the mirror. In the center of the field of view, as seen in the eye-piece, the thermo-couple is seen as a black spot, and this must be overlapped on all sides by the image of the hot body to obtain the correct temperature. It is interesting to observe that the distance of the telescope from the source of heat does not affect the reading of the

Fig. 59. — Sensitive Galvanometer of Fery Radiation Pyrometer.

Fig. 60. — Telescope of Fery Radiation Pyrometer.

instrument. When the telescope gets nearer the hot body the mirror **M** receives of course more heat, but at the same time this greater amount of heat is distributed over a larger image and the intensity of the heat remains the same.

Radiation pyrometers are calibrated in terms of the radiation from a so-called " black body," which is approximately realized by a uniformly heated enclosure. It is only for " black bodies," such as carbon, coal, etc., that the temperature is exactly proportional to the fourth root of the heat energy. Readings obtained when measuring the temperature of a body not inside a closed chamber with hot walls will in some cases be very much lower than the true temperature. For a piece of heated coal the error is very small due to lack of enclosure, while in the case of molten copper or tin with a clean surface the temperature reading may be 100 degrees Fahrenheit too low. Conditions as regards enclosure are,

however, satisfactory in most practical cases where the instrument is frequently used, such as taking the temperature of boiler furnaces, gas producers and retorts, annealing and hardening furnaces, etc. Error due to the furnace door being open for an instant when the observation is to be made is practically negligible, especially as these instruments are actually calibrated under this condition. If excess of air in a furnace is likely to reduce the temperature while sighting, a large tube of cast iron or fire-clay closed at the end toward the fire can be built into the furnace wall. By sighting through the open end upon the closed end which should be at the furnace temperature very satisfactory results are obtained.

Observations made with such pyrometers of incandescent bodies or gases do not give the true temperature. It is generally assumed, however, that they can be used to measure fairly accurately the temperature of heated chambers when focused upon the walls,[1] because of the reflection going on in all directions. In most cases the flame temperature can be taken the same as that of the surrounding walls.

A relatively large area is usually required to sight radiation pyrometers. It is stated that the distance from the telescope to the hot body can be as much as 30 times the diameter of the hot body and the telescope can be taken as much nearer as desired without changing the reading of instrument. Before taking observations the pointer of the galvanometer must be set at zero, the instrument receiving no heat rays during this adjustment. The **readings of temperature** made with such instruments are obviously the difference between the temperatures of the hot body and of the room.

Optical Pyrometers. Another type of pyrometer, based in principle upon the measurement of the brightness of the hot body by comparison with a standard lamp, is shown in **Fig. 61.** In order to use this instrument, known as Wanner's, the incandescent (osmium filament) lamp must first be standardized by comparison with an amylacetate oil lamp of constant candle power. Then after standardizing it is only necessary to focus the instrument upon the hot body to be measured and the temperature is read directly on the graduated scale at the eye-piece.

Temperature readings from optical pyrometers are actual and are not differences depending on the temperature of the room.

Both the Fery and Wanner pyrometers have a satisfactory range from 800 to 4000 degrees Fahrenheit. At the lower temperatures the average error of such instruments is about 3 degrees Fahrenheit and the maximum error at temperatures above 3000 has been shown to be not more than 20 degrees.

[1] "*Heat Energy and Fuels,*" by Hanns von Jüptner, page 76.

Furnace temperatures can be determined approximately from the values corresponding to the color of the fire. All temperatures are in degrees Fahrenheit.

Red — just visible	900	Orange	2000
Dull red	1250	White	2350
Cherry red	1600	Dazzling white	2700

Radiation and optical pyrometers are invaluable for determining the temperatures of the various parts of a furnace, of the walls of the setting of a steam boiler, of various portions of a bed of coals, etc. It is sometimes stated that an optical pyrometer is a means for measuring temperatures of objects " miles away."

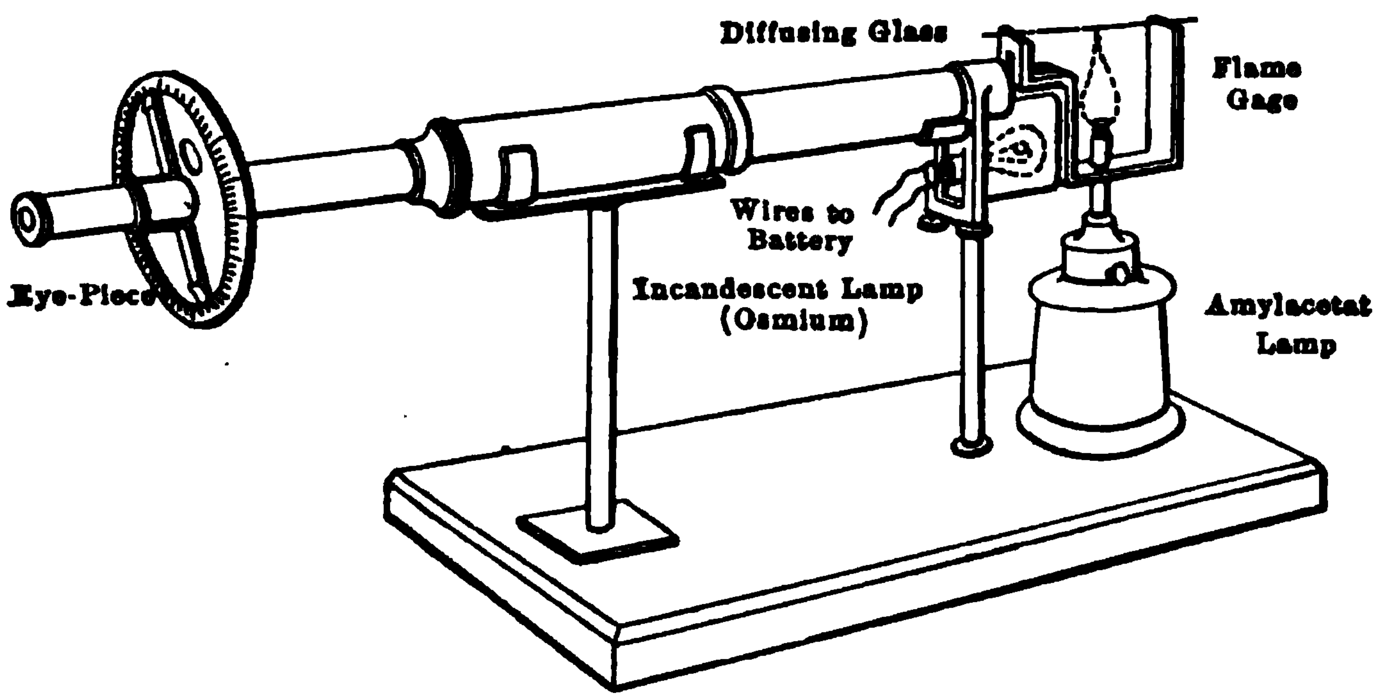


FIG. 61. — Wanner Optical Pyrometer in Position for Standardizing.

Calorimetric Pyrometers. If the specific heat and weight of a body are known, its temperature can be obtained by observing the rise in temperature of a known quantity of water into which the body is thrown.

More in detail the method consists in the determination of temperature by putting a ball of metal or other refractory material into the medium of which the temperature is to be measured. When the ball has become heated uniformly throughout its mass to the temperature of the medium it is transferred quickly to a cup heavily jacketed with non-conducting material in which there is a known weight of water at a known temperature. Copper, wrought iron and fire-clay are suitable materials. Specific heats of these materials at about 500 degrees Fahrenheit are respectively .097, .110 and .180. Since metals are readily attacked by furnace gases they should be protected when used in this way in a crucible of refractory material.

This method is often very serviceable in places or at times when accurate pyrometers are not available. On account of the " personal " error liable to enter, such determinations should be repeated several times to check the results. Calculations required are as follows.[1]

[1] A more complete description of calorimetric pyrometers and the precautions to be observed for accuracy will be found in *Transactions of the American Society of Mechanical Engineers*, vol. VI, page 712.

Let w_1 = weight of the ball, pounds.
w_2 = weight of the cup (only the "inner" vessel),[1] pounds.
w_3 = weight of the water in the cup, pounds.
t_1 = initial temperature of water, degrees Fahrenheit.
t_2 = final temperature of the water, degrees Fahrenheit.
t_0 = temperature of the heated ball, degrees Fahrenheit.
s_1 = specific heat of the ball.
s_2 = specific heat of the cup.

Then $$w_1s_1(t_0 - t_2) = (w_2s_2 + w_3)(t_2 - t_1),$$

$$t_0 = \frac{(w_2s_2 + w_3)(t_2 - t_1)}{w_1s_1} + t_2. \qquad (2)$$

Seger Pyrometer Cones. For many purposes when a pyrometer cannot be well placed fusible Seger cones are used. Such cones are made of several different oxides mixed in a manner to give a definitely known melting point for each one. The melting points range from 590 degrees to 1850 degrees Centigrade by steps of from 20 to 30 degrees, each having a standard number. These cones are carefully graded, so that if one has had some experience with them, temperatures can be estimated to about the nearest ten degrees in Centigrade. Four of these cones are shown in **Fig. 62.**

Fig. 62. — Seger Cones after Use.

When a series of cones is placed in a furnace the one having the lowest melting point begins to turn over first. The temperature corresponding to the cone number is reached when the tip of the cone has bent over and just touches the surface on which it is standing. Hence the highest temperature reached when the cones shown in the illustration were used was about half way between that corresponding to each of the two middle cones. According to the numbers on the cones the temperature,

[1] It would be more accurate, of course, to use in the calculation the water equivalent of the whole vessel, as is done in coal calorimetry. See page 210. Units given are in pounds and degrees Fahrenheit, but other units, provided they are corresponding, can be used in the equation given.

as given by the following table, was between 830 and 860 degrees Centigrade. The greatest disadvantage with this system is that there is no way of observing a decrease in the temperature, or, in other words, only the maximum temperature is indicated.

The following table gives the temperatures, in degrees **Centigrade,** at which the Seger cones will begin to melt:

Seger Cone No.	Temp. Deg. C.	Seger Cone No.	Temp. Deg. C.	Seger Cone No.	Temp. Deg. C.
022	590	04	1070	15	1430
021	620	03	1090	16	1450
020	650	02	1110	17	1470
019	680	01	1130	18	1490
018	710	1	1150	19	1510
017	740	2	1170	20	1530
016	770	3	1190	....	....
015	800	4	1210	26	1650
014	830	5	1230	27	1670
013	860	6	1250	28	1690
012	890	7	1270	29	1710
011	920	8	1290	30	1730
010	950	9	1310	31	1750
09	970	10	1330	32	1770
08	990	11	1350	33	1790
07	1010	12	1370	34	1810
06	1030	13	1390	35	1830
05	1050	14	1410	36	1850

Two types of mercury thermometers protected by heavy metal cases are illustrated by **Figs. 63** and **64.** It will be observed that a very satisfactory thermometer well is a part of the casing. The one shown in **Fig. 64** has graduations for reading both temperatures and pressures. A thermometer of this type is particularly useful in pipes carrying hot boiler feed-water. When the temperature is above 212 degrees Fahrenheit the thermometer will indicate that the water is being heated at a pressure higher than atmospheric. For water heated in closed vessels or pipes there is for every temperature a corresponding pressure as given in tables of the properties of saturated steam.[1]

Relative Accuracy of Thermometers and Pyrometers. For low-temperature work mercury thermometers are generally preferred as they can be made to almost any degree of accuracy required. For temperatures above 500 degrees Fahrenheit electric resistance thermometers and pyrometers come into use. When provided with a delicate galvanometer electric resistance thermometers can be used with a very high degree of accuracy, and in fact temperature differences can be determined with them very much more accurately than with the best mercury ther-

[1] Short and very much abbreviated tables of the properties of saturated steam are given in the Appendix.

mometers. Next in degree of accuracy are probably thermo-electric pyrometers; and it is interesting that a pyrometer of this kind can be readily made by twisting together at their ends, rods of wrought iron and

Fig. 63. — Combined Thermometer Well and Protective Casing.

Fig. 64. — Combined Thermometer and Pressure Gage for Boiler Feed-water Pipes.

nickel. It is not essential that the ends should be welded but welding (preferably electric) gives the couple greater permanency, by preventing the accumulation of dust interfering with electrical conductivity. The loose ends can be connected up to the binding posts of a millivoltmeter by insulated copper wires and calibrated. The only disadvantage will be that it will not have a scale reading directly in degrees of temperature. The wrought-iron and nickel rods should be covered with a winding of asbestos tape to keep them separated. Mechanical pyrometers are not very accurate. Optical and radiation pyrometers have a special field beyond the limits of the other types.

CHAPTER III

DETERMINATION OF THE MOISTURE IN STEAM

UNLESS the steam used in the power plant is superheated it is said to be either dry or wet, depending on whether or not it contains water in suspension. The general types of **steam calorimeters,** used to determine the amount of moisture in the steam, may be classified under three heads:

1. Throttling or superheating calorimeters.
2. Separating calorimeters.
3. Condensing calorimeters.

Throttling or Superheating Calorimeters. The type of steam calorimeter used most in engineering practice operates by passing a sample of the steam through a small orifice, in which it is superheated by throttling. A very satisfactory calorimeter of this kind can be made of pipe fittings as illustrated in **Fig. 65.** It consists of an orifice **O,** discharging into a chamber **C,** into which a thermometer **T** is inserted, and a mercury manometer is usually attached to the cock **V_3,** for observing the pressure in the calorimeter.

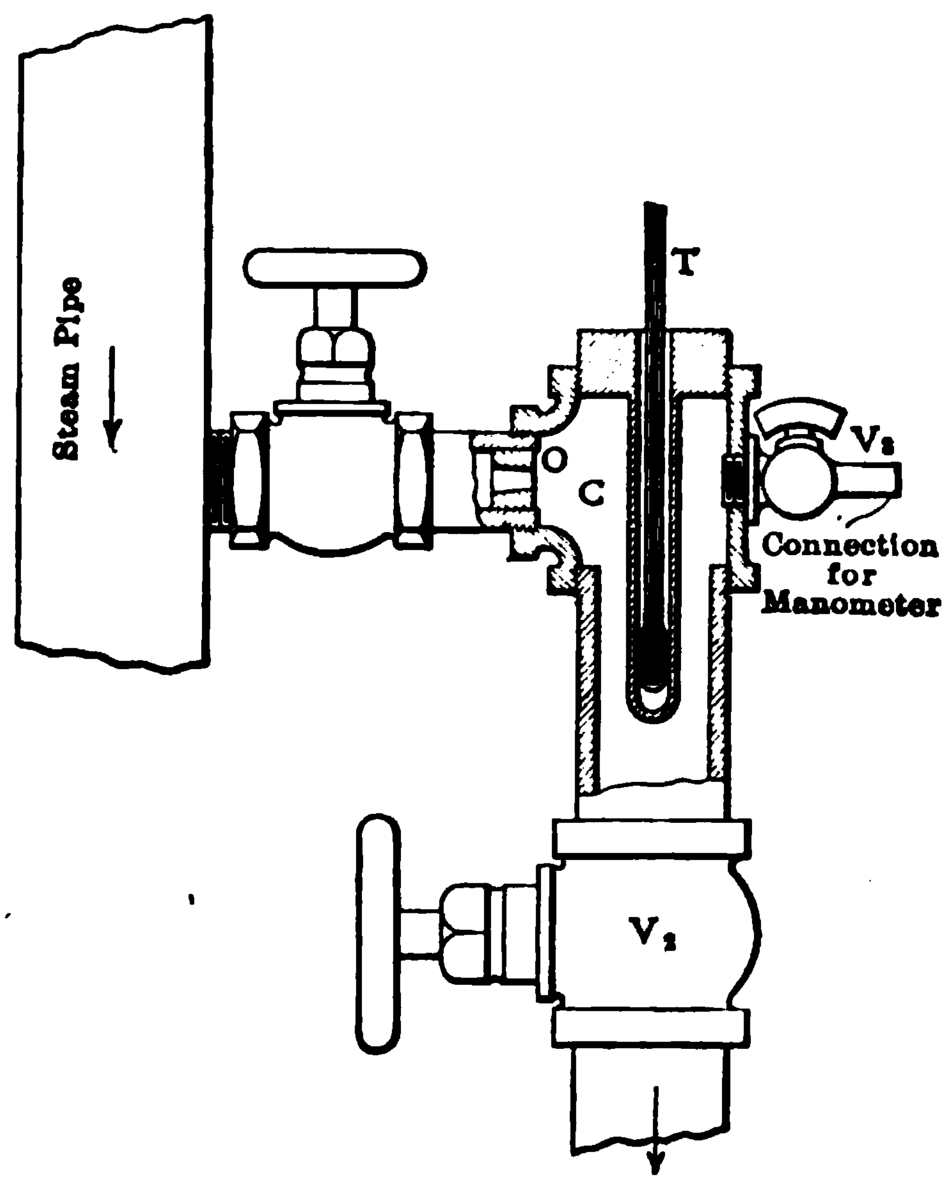


FIG. 65. — Simple Throttling Calorimeter.

It is most important that all parts of calorimeters of this type, as well as the connections leading to the main steam pipe, should be very **thoroughly lagged** by a covering of good insulating material. One of the best materials for this use is hair felt, and it is particularly well suited for covering the more or less temporary pipe fittings, valves, and nipples through which steam is brought to the calorimeter. Very many throttling calorimeters have been declared useless by engineers and put into the scrap heap merely because the small pipes leading to the calorimeters were not properly lagged, so that there was too much radiation, producing, of course, con-

densation, so that the calorimeter did not get a true sample. It is obvious that if the entering steam contains too much moisture the drying action due to the throttling in the orifice may not be sufficient to superheat. It may be stated in general that unless there is about 5 to 10 degrees Fahrenheit of superheat in the calorimeter, or in other words unless the temperature on the low-pressure side of the orifice is at least about 5 to 10 degrees Fahrenheit higher than that corresponding to the pressure in **the calorimeter,** there may be some doubt as to the accuracy of results.[1] The working limits of throttling calorimeters vary with the initial pressure of the steam. For 35 pounds per square inch absolute pressure the calorimeter ceases to superheat when the percentage of moisture exceeds about 2 per cent; for 150 pounds absolute pressure, when the moisture exceeds about 5 per cent; and for 250 pounds absolute pressure, when it is in excess of about 7 per cent. For any given pressure in the main the exact limit varies slightly, however, with the pressure in the calorimeter.

In connection with a report on the standardizing of engine tests, the American Society of Mechanical Engineers[2] published the following instructions regarding the method to be used for obtaining a fair sample of steam from the main pipes. It is recommended in this report that the calorimeter shall be connected with as short intermediate piping as possible with a so-called **calorimeter sampling nozzle** made of $\frac{1}{2}$-inch pipe and long enough to extend into the steam pipe "nearly across to" the opposite wall. The end of this nipple is to be closed so that the steam must enter through not less than twenty $\frac{1}{8}$-inch holes "equally distributed from end to end and preferably drilled in irregular or spiral rows, with the first hole not less than $\frac{1}{2}$-inch from the inner wall of the pipe."

"The sampling nozzle should not be placed near a point where water may pocket or where such water may affect the amount of moisture contained in the sample. Where non-return valves are used, or where there are horizontal connections leading from the boiler to a vertical outlet, water may collect at the lower end of the uptake pipe and be blown upward in a spray which will not be carried away by the steam owing to a

[1] The same general statement may be made as regards determinations of superheat in engine and turbine tests. Experience has shown that tests made with from 0 to 10 degrees Fahrenheit superheat are not reliable, and that the steam consumption in many cases is not consistent when compared with results obtained with wet or more highly superheated steam. The errors mentioned, when they occur, are probably due to the fact that in steam, indicating less than 10 degrees Fahrenheit superheat, water in the liquid state may be taken up in "slugs" and carried along without being entirely evaporated.

[2] *Transactions American Society of Mechanical Engineers*, vol. 21; and the *Journal*, Nov., 1912, pages 1713–14.

lack of velocity. A sample taken from the lower part of this pipe will show a greater amount of moisture than a true sample. With goose-neck connections a small amount of water may collect on the bottom of the pipe near the upper end where the inclination is such that the tendency to flow backward is ordinarily counterbalanced by the flow of steam forward over its surface; but when the velocity momentarily decreases the water flows back to the lower end of the goose-neck and increases the moisture at that point, making it an undesirable location for sampling. In any case it should be borne in mind that with low velocities the tendency is for drops of entrained water to settle to the bottom of the pipe, and to be temporarily broken up into spray whenever an abrupt bend or other disturbance is met."

If it is necessary to attach the sampling nozzle at a point near the end of a long horizontal run, a drip pipe should be provided a short distance in front of the nozzle, preferably at a pocket formed by some fitting, and the water running along the bottom of the main drawn off, weighed, and added to the moisture shown by the calorimeter, or better, a steam separator should be installed at the point noted.

In testing a boiler the sampling nozzle should be located as near as possible to the boiler, and the same is true as regards the thermometer well when the steam is superheated. In a turbine or engine test these locations should be as near as practicable to the throttle valve. In the test of a plant where it is desired to get complete information, especially where the steam main is unusually long, sampling nozzles or thermometer wells should be located at both the boiler and the engine, so as to obtain as complete data as may be required. The sample of steam should always be taken from a **vertical pipe** as near as possible to the engine, turbine, or boiler being tested. Good examples of calorimeter nipples are illustrated in **Figs. 67** and **74**.

Never close and usually do not attempt to adjust the discharge valve $\mathbf{V_2}$ (**Fig. 65**) without first closing the gage cock $\mathbf{V_3}$. Unless this precaution is taken the pressure may be suddenly increased in the chamber **C**, so that if a manometer is used the mercury will be blown out of it, and if, on the other hand, a low-pressure steam gage is used it may be ruined by exposing it to a pressure much beyond its scale.

Usually it is a safe rule to begin to take observations of temperature in calorimeters after the thermometers have indicated a maximum value and have again receded slightly from this maximum.

The quality or relative dryness of wet steam is easily calculated by the following method. Using the symbols,

$\mathbf{p_1}$ = steam pressure in main, lbs. per sq. in. abs.

$\mathbf{p_2}$ = steam pressure in calorimeter, lbs. per sq. in. abs.

$\mathbf{t_c}$ = temperature in calorimeter, deg. Fahr.

r_1 and q_1 = heat of vaporization, and heat of liquid corresponding to pressure p_1, B.t.u. per pound.
H_2 and t_2 = total heat (B.t.u.) and temperature (deg. Fahr.) corresponding to pressure p_2.
c_p = specific heat of superheated steam. Assume 0.46 for low pressures existing in calorimeters.[1]
x_1 = initial quality of steam (a decimal).
$100 (1 - x_1)$ = initial moisture in steam, per cent.

Total heat in a pound of wet steam flowing into the orifice is

$$x_1 r_1 + q_1,$$

and after expansion, assuming all the moisture is evaporated, the total heat of the same weight of steam is

$$H_2 + c_p (t_c - t_2).$$

Then assuming no heat losses and putting for c_p its value 0.46 we have,

$$x_1 r_1 + q_1 = H_2 + 0.46 (t_c - t_2), \qquad (3)$$

or

$$x_1 = \frac{H_2 + 0.46 (t_c - t_2) - q_1}{r_1}. \qquad (3')$$

Charts for Moisture Determinations. A small section of the total-heat-entropy chart as provided in modern steam tables is shown in **Fig. 66.** It is arranged particularly for determinations of the quality of steam with a throttling calorimeter without using the equations above. Horizontal lines in the chart are those of constant total heat of the steam, and represent the process in a throttling calorimeter. To illustrate the application of the chart let the initial pressure of steam be 165 pounds per square inch absolute and the reading of the thermometer on the low-pressure side of the calorimeter be 270 degrees Fahrenheit. The pressure in the calorimeter is 15.2 pounds per square inch absolute. To find the quality x start at the intersection of the temperature line for 270 degrees with the 15.2 pounds pressure line and go across the chart horizontally to the 165 pounds line, then the "lines of constant quality" indicate that the quality of the steam is 0.979.

When a U-tube manometer is used to determine the pressure in a calorimeter of the type illustrated in **Fig. 65,** this pressure can be obtained very accurately, and an excellent means is provided for **calibrating the thermometer** in the calorimeter just as it is to be used. The calibration would be made, of course, by the method of comparing with the temperature corresponding to known pressures explained on page 34. In order to avoid having superheated steam in the calorimeter for this calibration

[1] Average values for the specific heat of superheated steam for any pressures and temperatures are given on page 309.

the felt or similar material usually needed for covering the valves and nipples between the main steam pipe and the calorimeter should be kept saturated with cold water.

The Barrus Throttling Calorimeter. An important variation from the type of throttling calorimeter shown in **Fig. 65** has been introduced quite widely by Mr. George H. Barrus. In this apparatus the **temperature** of the steam admitted to the calorimeter is observed instead of the pressure and a very free exhaust is provided so that the pressure in the calorimeter is atmospheric. This arrangement simplifies very much

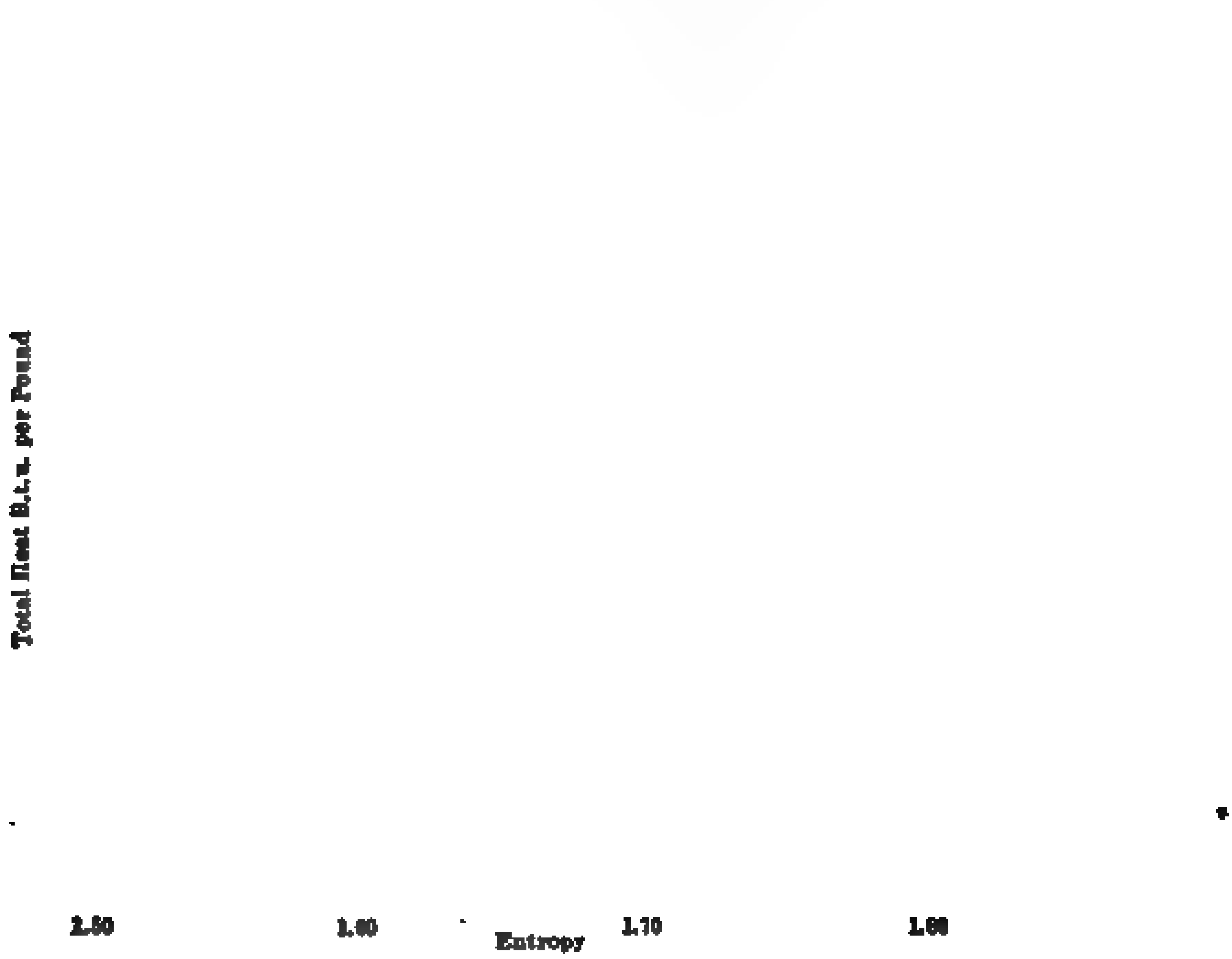


FIG. 66. — Chart for Determining Quality of Steam with any Throttling Calorimeter.

the observations to be taken, as the quality of the steam x_1 can be calculated by equation (3′) by observing only the two temperatures t_1 and t_c, taken respectively on the high- and low-pressure sides of the orifice in the calorimeter. This calorimeter is illustrated in **Fig. 67**. The two thermometers required are shown in the figure. Arrows indicate the path of the steam.[1]

The orifice in such calorimeters is usually made about $\frac{3}{32}$ inch in diameter; and for this size of orifice the weight of steam[2] discharged per

[1] *Transactions American Society of Mechanical Engineers*, vol. 11, page 790.

[2] Formulas for calculating the exact weight of steam discharged from a nozzle are given on pages 189 and 190. In boiler-tests corrections should be made for the steam

hour at 175 pounds per square inch absolute pressure is about 60 pounds. It is important that the orifice should always be kept clean, because if it becomes obstructed there will be a reduced quantity of steam passing through the instrument, making the error due to radiation relatively more important.

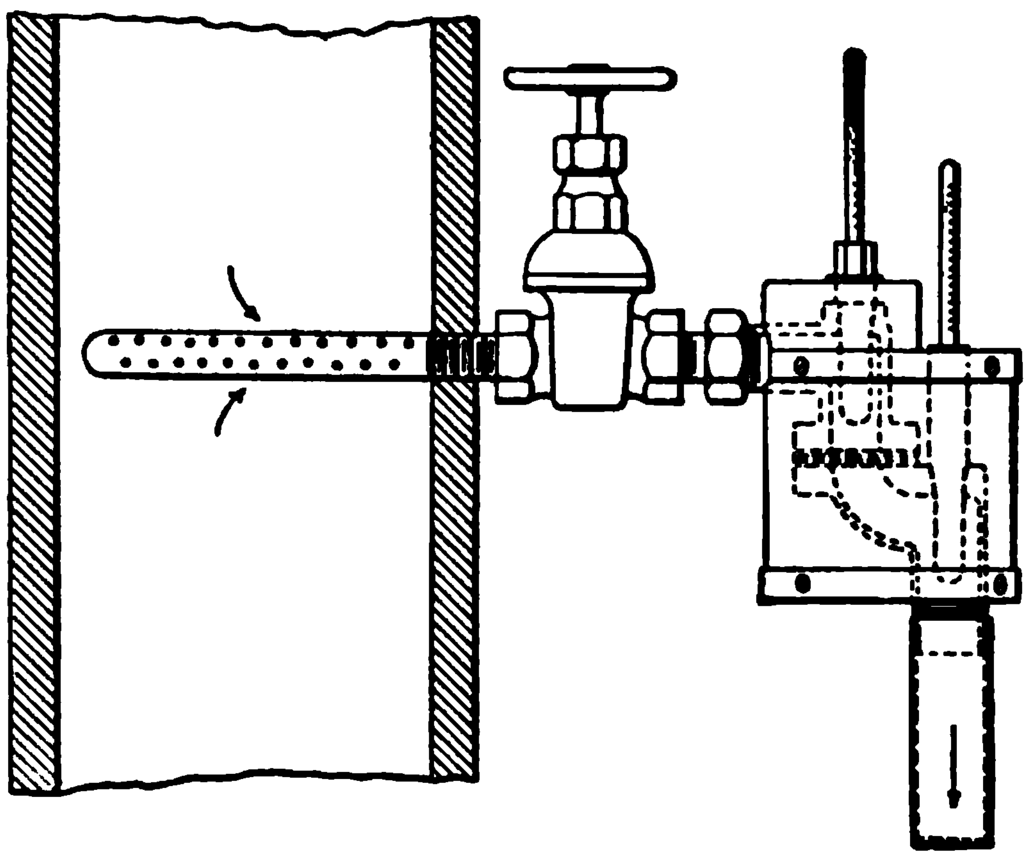


FIG. 67. — Barrus Throttling Steam Calorimeter.

In order to free the orifice from dirt or other obstructions the connecting pipe or calorimeter nipple to be used for attaching the calorimeter to the main steam pipe should be blown out thoroughly with steam before the calorimeter is put in place. The connecting pipe and valve should be covered with hair felting not less than $\frac{3}{4}$ inch thick. It is desirable also that there should be no leak at any point about the apparatus, either in the stuffing box of the supply valve, the pipe joints, or in the union.

Fig. 68 is a diagram for the determination of the quality of steam which is particularly suitable for use in connection with calorimeters of the Barrus type.

Abscissas in this diagram are temperatures in the calorimeter t_c, and the ordinates are the initial temperatures t_1 of the steam before expansion in the calorimeter.

With the help of such a diagram the Barrus calorimeter is particularly well suited for use in power plants, where the quality of the steam is entered regularly on the log sheets. The percentage of moisture is obtained immediately from two observations without any calculations.

A very good design of throttling calorimeter recommended by the Power Test Committee of the A.S.M.E. to be accepted as the standard for tests is shown in **Fig. 69.** The calorimeter is made practically throughout of $\frac{1}{2}$-inch pipe fittings and has an orifice $\frac{5}{64}$ inch in diameter in a flat plate. (**Fig. 70.**) This orifice is of a suitable size to throttle steam at the usual boiler pressures down to atmospheric. The wooden box should be filled with hair felt, 85 per cent magnesia, or an equally

discharged from the steam calorimeters. The Power Test Committee of the A.S.M.E. suggest the use of Napier's formula, believing it to be sufficiently accurate for this kind of work.

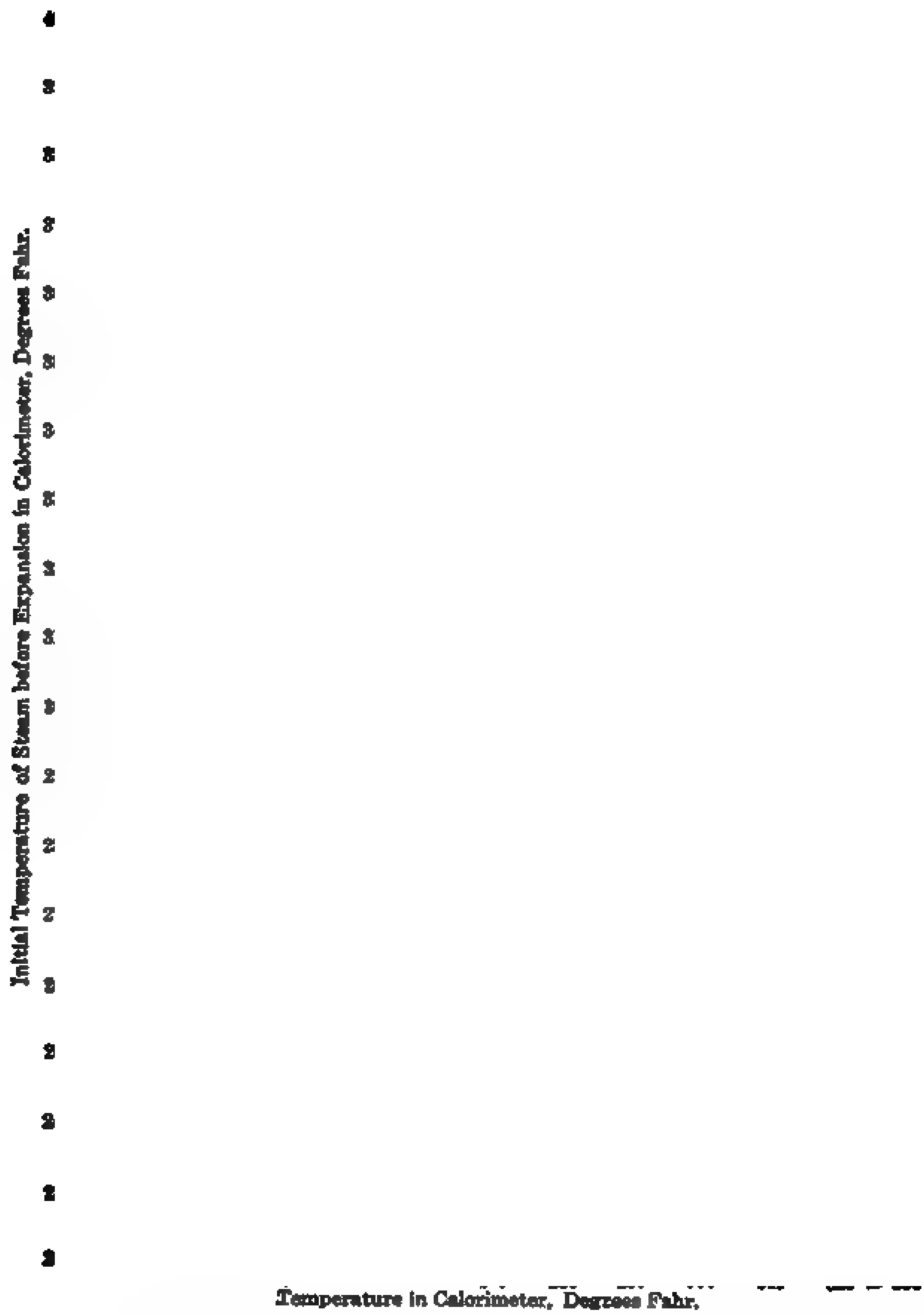


FIG. 68.—Chart for Determining Quality of Steam from Temperature Observations. (Atmospheric Pressure in Calorimeter.)

good heat insulator. This committee states the following method of calculating the quality of steam with this instrument:

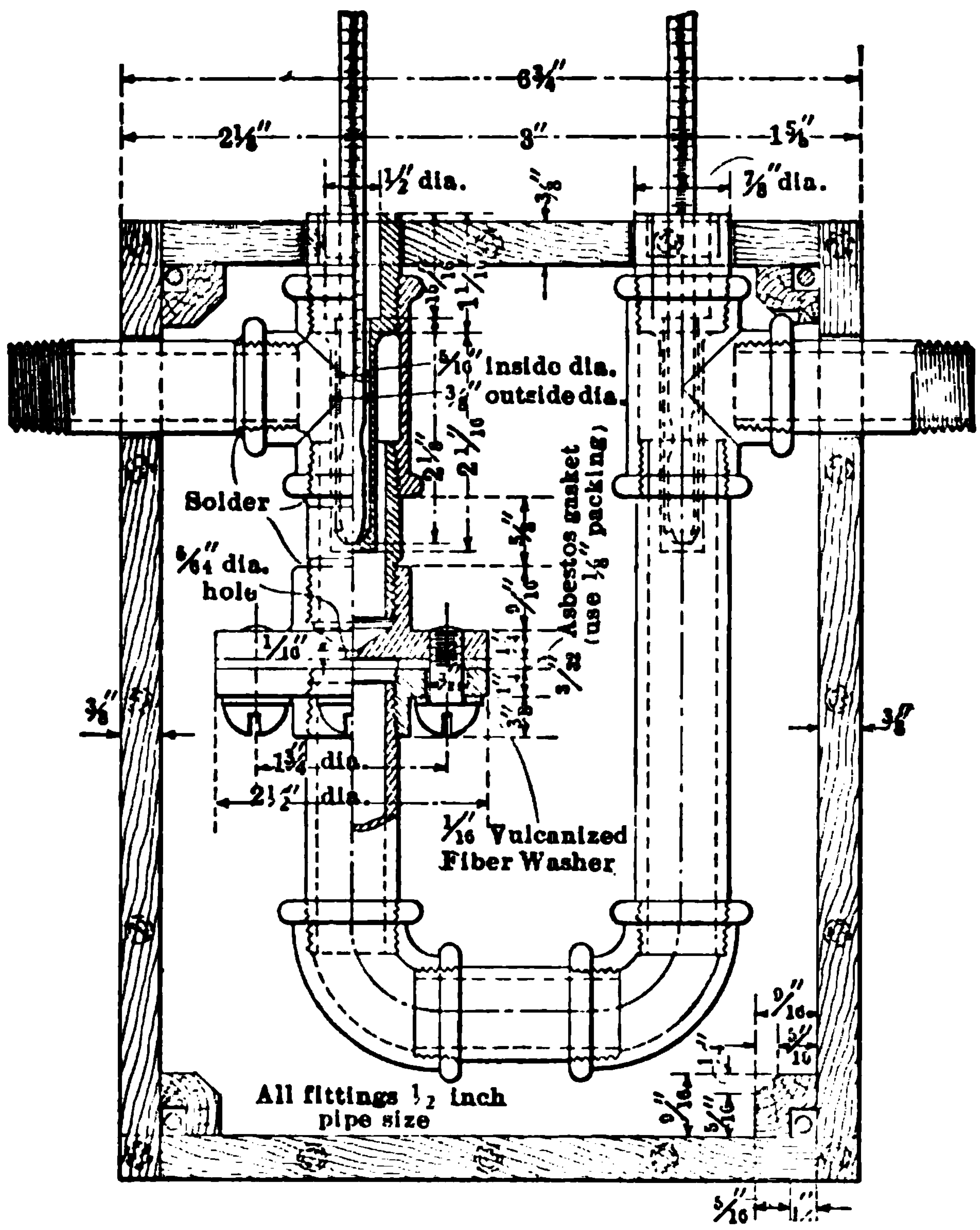


FIG. 69. — Simple Throttling Calorimeter. (A.S.M.E.Type.)

"The percentage of moisture is determined by observing the number of degrees of cooling that the thermometer in the low-pressure steam shows as its 'normal' reading for dry steam and dividing that number by the 'constant' number of degrees representing one per cent of moisture. To determine this 'normal' reading corresponding to dry steam the instrument should be attached to a horizontal steam pipe in such a way that the sampling nozzle projects upward to near the top of the pipe, there being no perforations, and the steam enters through the open top of the nozzle. The test should be made when the steam in the pipe is in a quiescent state, and

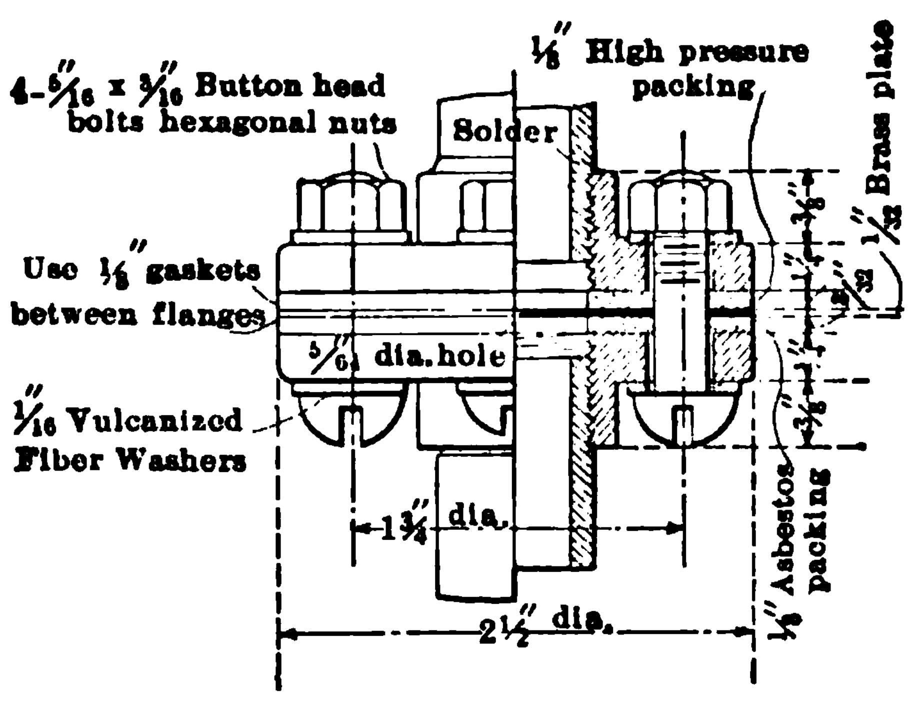


FIG. 70. — Detail of Orifice for FIG. 69.

the steam pressure is maintained constantly at the point observed on the main trial. If the steam pressure falls during the time the observations are being made, the test should be continued long enough to obtain the effect of an equivalent rise of pressure.

"To find the constant for one per cent of moisture divide the latent heat of the steam supplied to the calorimeter at the observed pressure or temperature by the specific heat of superheated steam at atmospheric pressure (0.46) and divide the quotient by 100. Finally ascertain the percentage of moisture by dividing the number of degrees of cooling by the 'constant' as above noted."

Separating Calorimeters. It was explained on page 56 that throttling calorimeters cannot be used for the determination of the quality of steam when for comparatively low pressures the moisture is in excess of 2 per cent, and when for average boiler pressures in modern engineering practice it exceeds 5 per cent. For higher percentages of moisture than these low limits separating calorimeters are most generally used. In these instruments the water is removed from the sample of steam by mechanical separation just as it is done in the ordinary steam separator installed in the steam mains of a power plant. There is provided, of course, a device for determining, while the calorimeter is in operation, usually by means of a calibrated gage glass, the amount of moisture collected. This mechanical separation depends for its action on changing very abruptly the direction of flow and reducing the velocity of the wet steam. Then since the moisture (water) is nearly 300 times as heavy as steam at the usual pressures delivered to the engine, the moisture will be deposited because of its greater inertia.

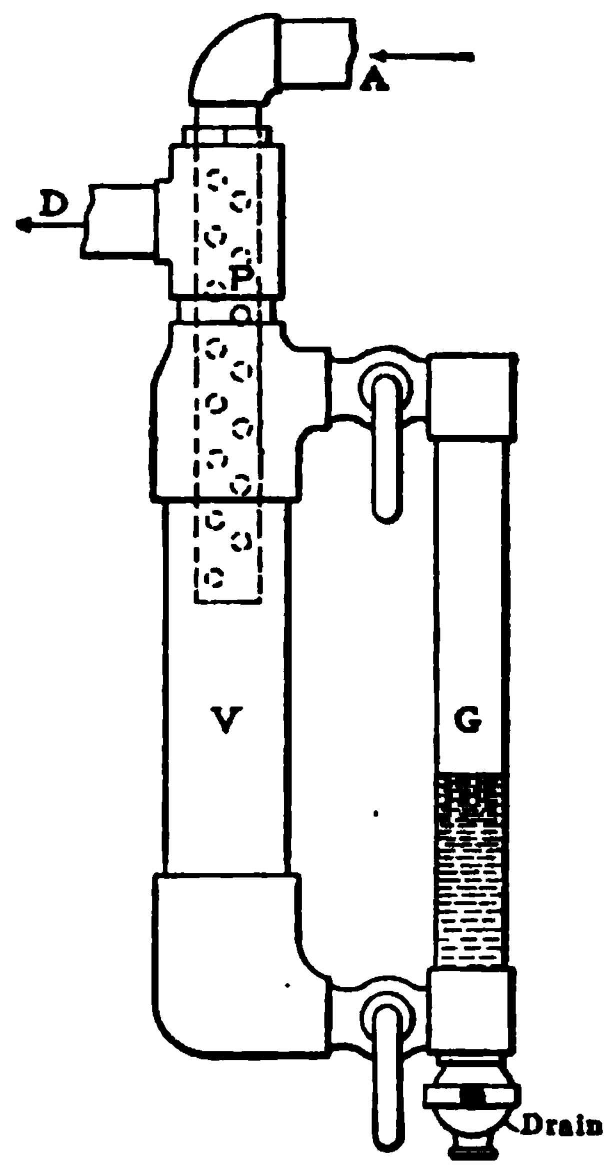


FIG. 71. — A Simple Separating Steam Calorimeter made of Pipe Fittings.

One of the simplest forms of separating calorimeters, made of pipe fittings, is shown in **Fig. 71.** Steam enters at **A**, passes down through the vertical pipe **P**, plugged at the lower end, from which it escapes through a large number of $\frac{1}{8}$-inch holes indicated in the figure. In passing through these holes the direction of flow is changed very abruptly, since the steam must go upward to be discharged at **D**. Moisture is deposited at the bottom of the vessel **V**, and its volume or weight can be determined from the height of the water in the gage glass **G** if the vessel has been calibrated. Steam discharged from **D** must be condensed and weighed in a pail or barrel containing cold water. The percentage of moisture is then found

by dividing the weight of water collected in the vessel **V** by the sum of the weight of steam condensed and the weight of water collected in **V**. This sum is, of course, the weight of the **wet** steam.

Radiation Loss. As in all calorimetry work, in order to obtain accurate results there should be a covering of hair felt $\frac{3}{4}$ inch thick over all parts of the apparatus, and even then the radiation loss is sometimes large enough to make corrections necessary. This correction is determined by operating two calorimeters which are exactly alike in construction and in the amount of felt covering, in series, and so arranged that the second takes the discharge of the first. If it is known that the discharge from the first calorimeter is perfectly dry steam[1] then the moisture collected in the second calorimeter is the condensation due to its own radiating surface, which should be the same as for the first. Calculations for corrected moisture determinations are made then by subtracting from the moisture collected in the first instrument the amount condensed in the second. When, of course, the radiation loss has been once determined it is not necessary to operate the second calorimeter.

FIG. 72. — Separating Steam Calorimeter with Steam Jacket.

Fig. 72 illustrates a form of separating calorimeter in which the improvement over the one shown in **Fig. 71** is in the addition of a steam-jacketing space receiving live steam at the same temperature as the sample. Steam is supplied through a pipe **A**, discharging into a cup **B**. Here the direction of the flow is changed through nearly 180 degrees,

[1] A small throttling calorimeter can be attached to the discharge from the first separating calorimeter to determine by a separate test whether or not the steam discharged is dry.

causing the moisture to be thrown outward through the meshes of the cup into the vessel **V**. The dry steam passes upward through the spaces between the webs **W** into the top of the outside jacketing chamber **J**, and is finally discharged from the bottom of this steam jacket through the nozzle **N**. This nozzle is considerably smaller than any other section through which the steam flows, so that there is no appreciable difference between the pressures in the calorimeter proper and the jacket. The scale opposite the gage glass **G** is graduated to show in **hundredths of a pound,** at the temperature corresponding to steam at ordinary working pressures, the variation of the level of the water accumulating. A steam pressure gage **P** indicates the pressure in the jacket **J**, and since the flow of steam through the nozzle **N** is roughly proportional to the pressure (see page 189), another scale in addition to the one reading pressures is provided at the outer edge of the dial. A petcock **C** is used for draining the water from the instrument, and by weighing the water collected corresponding to a given difference in the level in the gage **G**, the scale opposite it can be readily calibrated. Too much reliance should not be placed on the readings for the flow of steam as indicated by the gage **P**, unless it is frequently calibrated. Usually it is very little trouble to connect a tube to the nozzle **N** and condense the steam discharged in a large pail nearly filled with water. When a test for quality is to be made by this method the pail nearly filled with cold water is carefully weighed and then at the moment when the level of the water in the water gage **G** has been observed the tube attached to the nozzle **N** is immediately placed under the surface of the water in the pail. The test should be stopped before the water gets so hot that some weight is lost by "steaming." The gage **P** is generally calibrated to read pounds of steam flowing in ten minutes. For the best accuracy it is desirable to use a pail with a tightly fitting cover into which a hole just the size of the tube has been cut.

Combined Separating and Throttling Calorimeters. Calorimeters as already described are effective in removing practically all the moisture in steam when the pressure is not lower than 25 pounds by gage. For lower pressures, particularly around atmospheric, recent experiments show that the efficiency of such calorimeters is in some cases not more than 80 per cent.[1] For this reason in the best current practice for determinations of moisture in low-pressure steam a throttling calorimeter is attached to the discharge of the separating calorimeter. Then if the separating calorimeter has been carefully calibrated for radiation loss and the steam escaping from the separating calorimeter is tested

[1] *Transactions American Society of Mechanical Engineers*, vol. 32 (1910), page 1132. The efficiency of the calorimeter is the ratio of the percentage of moisture taken out by the separating calorimeter to the total percentage of moisture.

again in a throttling instrument, it is possible to make correct determinations for the percentage of moisture in the steam of almost any degree of wetness. An apparatus of this kind which is reported to have done excellent service in tests of very large low-pressure steam turbines, operating with the exhaust from reciprocating steam engines in New York city, is shown in Fig. 73. The most unique feature of this apparatus is the sampling tube. It was found that for this low-pressure steam the ordinary sampling tube of perforated pipe (see Figs. 67 and 74) did not

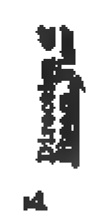


Fig. 73. — Combined Separating and Throttling Steam Calorimeter.

give a reliable sample. It was also found necessary that the sample should be taken from the main without changing its direction or velocity until it is actually inside the sampling pipe. If the direction of flow of wet steam is suddenly changed when entering the sampling nozzle, the entrained moisture, because of its greater specific gravity on the one hand and the very slight skin friction between it and the surrounding dry steam on the other, will cause it to continue in its path in a straight line, so that there is a tendency for only dry steam to enter the nozzle. Also if the velocity of the steam in the sampler is greater than that in the

main, there is a tendency for the dry steam to "accelerate" into the nozzle, leaving the moisture behind. It has been stated that this action has not been observed in tests of steam at high pressures, because (1) of smaller differences between the specific gravity of high-pressure steam and water; (2) greater skin friction; (3) the highly divided state of the moisture.[1]

As the throttling calorimeter is ordinarily used it would have very little capacity when used with steam pressures only a little above atmospheric; but by making it discharge into a receiver in which a vacuum of about 28 inches of mercury is maintained the throttling portion of the calorimeter will evaporate from two to three per cent of moisture.

The apparatus shown in **Fig. 73** consists of the $\frac{3}{8}$-inch brass nozzle on the sampling tube which is bent to point in the direction opposite to the flow of the steam. The lip of this nozzle is filed to a knife-edge to avoid disturbing the current of steam around the mouth of the sampler by eddies and impact against a thick lip. This sampling tube is set up so that it extends into the main steam pipe one-sixth of the diameter of the pipe, where it has been observed to give practically a true average sample. When in operation the valve at the sampling tube is opened wide and the flow is regulated by means of the lever cock between the separating and the throttling calorimeters. The necessary throttling action ordinarily produced by an orifice is produced by this cock. A vacuum is maintained in the throttling portion from which the discharge is carried to a small cooling receiver in which the steam is condensed. From this receiver it flows to a "volumetric" measuring tank of which the top is tightly closed and connected by $\frac{1}{4}$-inch pipes to the main condenser. A spy-glass shown at the left in the figure is useful for proving that the calorimeter is working properly. It often happens that when the superheat in the calorimeter is less than from five to eight degrees Fahrenheit there is some moisture passing through and the spy-glass will invariably show it. As the spy-glass is most conveniently made of $\frac{3}{4}$-inch gage glass its area is not large enough to carry all the steam, and a by-pass connection is arranged as shown. The large size of the parts is necessary on account of the very large specific volume of the low-pressure steam.

All parts of the apparatus are carefully covered with magnesia-asbestos covering 2 inches thick. For the normal rate of flow for the instrument, the radiation can be made less than 0.1 per cent.

Calculation of Percentage Moisture for Combination Separating and Throttling Calorimeter. Quality of steam x_1 is calculated for a combination calorimeter as follows:

[1] H. G. Stott, *Transactions American Society of Mechanical Engineers*, vol. 32 (1910), page 77.

Let w_1 = weight of moisture collected in the separating calorimeter in a given time, in pounds.

w_2 = weight of dry steam condensed after passing through the throttling calorimeter, in pounds.

x_2 = quality of steam discharged from separating portion as determined by the throttling calorimeter;

then without sensible error,

$$1 - x_1 = \frac{w_1}{w_1 + w_2} + (1 - x_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

and in terms of " quality " (always a decimal), we have more accurately,[1]

$$x_1 = x_2 \times \frac{w_2}{w_1 + w_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4')$$

Still another type of combined calorimeter is illustrated in Fig. 74. In this instrument the sample of steam is collected by the perforated tube in the main steam pipe. The temperature before expansion in the throttling plug is indicated by the thermometer marked T_1, and another thermometer T_2 gives the temperature after throttling. A scale S opposite the glass water gage G is used to show the weight of water separated from the steam. The proportion of moisture separated in relation to the total weight of wet steam passing through the instrument is the percentage of moisture separated. This percentage added to the percentage of moisture determined by throttling as calculated from the readings of the thermometers gives the approximate total percentage of moisture, as by equation (4) above.

A calorimeter of this type is particularly useful for making tests in power plants where the quality of the steam may vary considerably. It sometimes happens that a test is started with nearly dry steam; but after a while something goes wrong in the boiler room, as for example too much water may be fed into the boilers causing " priming," and the throttling calorimeter becomes useless for determining the quality. If a separating calorimeter is at hand this may be substituted; but to make a change often takes some time and meanwhile some observations may be lost. If, however, one of these combined calorimeters is used, it will operate satisfactorily as a simple throttling instrument when the steam is nearly dry; and, without adjustment, will also take care of very wet steam.

This type is also very useful to consulting engineers for making tests

[1] If the radiation test shows it is large enough to be appreciable, and if R is weight of condensation due to radiation in pounds in a given time corresponding to that for the other units, then

$$x_1 = x_2 \times \frac{w_2 + R}{w_1 + w_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4'')$$

at plants where there is no definite information available as to the quality of steam supplied by the boilers.[1]

Electric Steam Calorimeters. For use with very wet steam the Thomas electric calorimeter, Fig. 75, has been designed. It consists essentially of a cylindrical vessel B containing a series of resistance coils of German silver wire for heating steam by means of the electric current passing through them. These coils are connected to the electric terminals or binding-posts shown in the figure, and are supported in a soapstone cylinder in which there are a large number of ¼-inch holes through which the coils pass up and down.

Fig. 74. — Ellison's Steam Calorimeter.

Fig. 75. — Thomas' Electric Steam Calorimeter.

Steam enters at the bottom of the vessel at A and passing upward through the heated coils the moisture contained in it is evaporated. The steam then passes up through a perforated casing filled with copper gauze and escapes through the pipe discharging at the side at C. A part of this latter pipe is made of a glass tube for observing the condition of the steam. A thermometer is inserted at T for observing the temperature after this reheating.

In the operation of the instrument a steady flow of steam must first be secured, then the electric current should be turned on to dry the steam.

[1] If the calorimeter is not exceptionally well lagged with a steam jacket or with heavy heat-insulating material, a correction for radiation is necessary, as explained for a separating calorimeter on page 64.

When it is dry or superheated the fog will disappear from the glass observation tube. The first step is to superheat the steam to some temperature **T**, requiring an electrical input of $\mathbf{E}_t$ watts. Then decrease the current until the steam is just dry, requiring an input of $\mathbf{E}_0$ watts. Then $\mathbf{E}_t - \mathbf{E}_0 = \mathbf{E}'$ which represents the watts required to superheat the steam **t** degrees to the temperature **T**. By applying a constant **K**, determined by experiments for a series of pressures and degrees of superheat, the following equation is obtained:

$$\mathbf{h} = \frac{\mathbf{KE}_0}{\mathbf{E}'},$$

where **h** is the heat (B.t.u.) required to dry a pound of steam. A series of curves giving values of **K** are supplied with each instrument. If **r** is the heat of vaporization of the steam corresponding to the pressure then the quality

$$\mathbf{x} = \frac{\mathbf{r} - \mathbf{h}}{\mathbf{r}}. \qquad (4a)$$

Although this apparatus is used for steam of high quality as well as low, it has not been generally used probably because throttling calorimeters are preferred on account of greater simplicity in both construction and operation, and because very often a source of electric current is not conveniently available where tests are to be made. No data are available comparing its efficiency with that of the combined separating and throttling calorimeters described in the preceding paragraphs, but for **accurate** tests the latter are generally preferred by engineers.

Errors in this type of instrument are likely to be due to a variable weight of steam discharging in a unit of time; that is, the weight discharged will be less for superheated than for dry saturated steam. If, however, the steam discharged is condensed and weighed, the error from this source can be eliminated. It is practically impossible to make enough curves of values of **K** for all the variables.

Barrel Calorimeters. There is still another kind of steam calorimeter, known as the Hoadley barrel type, deserving some attention. It is one of the oldest forms of apparatus for making determinations of the quality of steam. In the classification made at the beginning of this chapter it belongs in the group of **condensing** calorimeters. Even with expert manipulations, ordinarily it is much less accurate than any of the calorimeters already described. A typical apparatus of this kind is shown in **Fig. 76.** It consists usually of a weighing barrel **B**, made of three concentric vessels of galvanized iron with the two annular spaces between the inner and outer vessels filled with pressed sheet cork or hair felt to reduce radiation to a minimum. It is usually arranged so that when the inner vessel has been nearly filled with water from the barrel **A**, a

quantity of the steam to be tested can be passed into it. The steam is admitted into the barrel in the most common forms by disconnecting the water hose **R** at **C** and making a temporary connection from the steam pipe out of which a sample is to be taken to a vertical pipe in the barrel of sufficient length to extend nearly to the bottom of the inner vessel. The pipe may be plugged at the lower end and sufficient area for the escape of steam is then secured by drilling into the pipe a number of $\frac{1}{8}$-inch holes for some distance from the lower end. This arrangement will make it easier to secure an equal rise in temperature in the different parts of the barrel. A float is usually provided to show the depth of water in the barrel, and a suitable stirring device or agitator consisting of paddles attached to a vertical shaft is also needed. This agitator when revolved stirs up the water and brings it to a constant temperature.

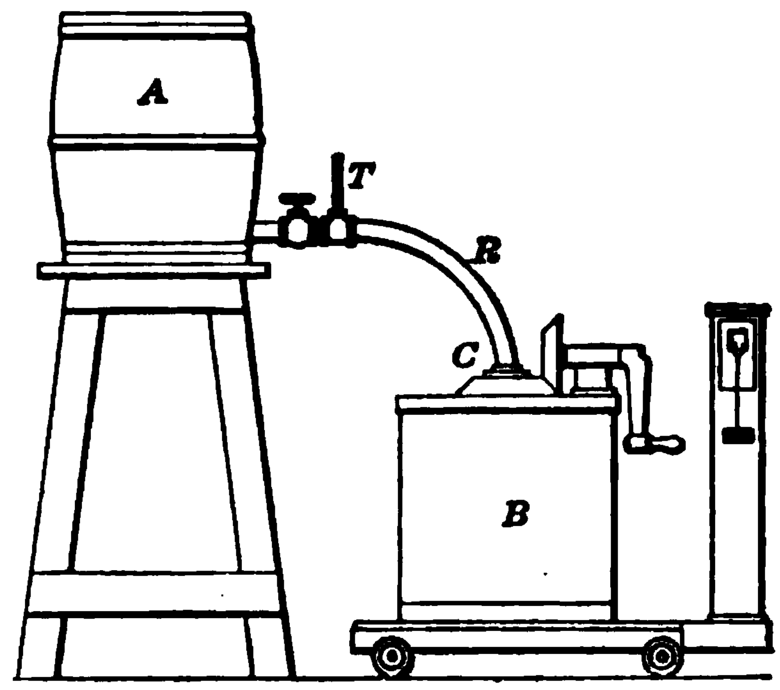


FIG. 76. — "Barrel" Steam Calorimeter.

Briefly the method to be pursued in the operation of the barrel calorimeter may be outlined as follows: First fill the barrel with cold water until the float shows that the water level is about six inches from the top. Then stir well, observe the temperature accurately and weigh carefully on a platform scales. The steam pipe should then be connected up to discharge into the water after first allowing the steam to blow off into the air, for the purpose not only of removing the condensation in the piping, but also to heat it to as nearly as possible the temperature of the steam. When the temperature has risen to about 120 degrees Fahrenheit the steam should be shut off and another weighing made to determine the amount of steam added. While the weighing is being done the water should be stirred vigorously and the highest temperature observed. For all the weighings the piping must be in exactly the same position as regards the connection on the barrel and all the pressure in the pipes must be relieved. When the piping between the calorimeter and the steam supply is connected by pipe fitters' unions these should be disconnected to insure the best accuracy. When, however, the connection is made by means of flexible rubber hose the weight can probably be obtained accurately enough without disconnecting the piping if the precaution is taken to relieve the pressure in the piping by opening a petcock located in the steam pipe near the point where it enters the barrel.

If, just before making the test for the quality of the steam, the cal-

orimeter is filled with water, heated with steam or otherwise to about 150 degrees Fahrenheit and again carefully drained, the barrel will be near its average temperature during the test, and no correction need probably be made for the heat absorbed by the calorimeter. In most cases it is preferable, however, to determine accurately the heat absorbed by the calorimeter and then make the proper corrections; but unless the work be done very carefully it is valueless. This correction is usually made by calculating the **water equivalent** or the capacity of the calorimeter to absorb heat measured by the similar capacity of water. The water equivalent is to be added to the weight of water in the calorimeter. Using then the following symbols:

w' = weight of water in calorimeter, in lbs.
w'' = weight of water added, in lbs.
t' = temperature of water in calorimeter, deg. Fahr.
t'' = temperature of water added, deg. Fahr.
t''' = temperature of mixture, deg. Fahr.
k = water equivalent, in lbs. Then

$$(w' + k)(t''' - t') = w''(t'' - t'''),$$

$$k = \frac{w''(t'' - t''')}{t''' - t'} - w'. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The temperature of the water added should be taken just as it **enters** the calorimeter and as near to it as possible.

The quality of the steam (x_0) to be determined is calculated as follows:

w_1 = weight of water in calorimeter, in lbs.
w_2 = weight of steam added, in lbs.
k = water equivalent of calorimeter, in lbs.
t_1 = initial temperature of water in calorimeter, deg. Fahr.
t_2 = final temperature of water, deg. Fahr.
p_0 = pressure of steam, lbs. per sq. in.
r_0 = heat of vaporization of steam (B.t.u.) corresponding to p_0.
q_0 and q_2 = sensible heat of steam (B.t.u.) corresponding to p_0 and t_2.

Then equating the heat lost by the steam to the heat gained by the water,[1]

$$(x_0 r_0 + q_0 - q_2)\, w_2 = (w_1 + k)(t_2 - t_1) \quad . \quad . \quad . \quad (6)$$

$$x_0 = \frac{(w_1 + k)(t_2 - t_1)}{w_2 r_0} + \frac{q_2 - q_0}{r_0}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

[1] Equation (7) can be made a little simpler for calculations without appreciable error by writing for $(q_2 - q_0)$ the difference between the corresponding temperatures $(t_2 - t_0)$.

The accuracy of this instrument depends principally on the care with which the various temperatures and the weight of the condensed steam are obtained. Usually it is very difficult to obtain accurately the temperature of the mixtures of water and steam. It is not unusual for determinations of moisture with such a calorimeter to vary for the same quality of steam and with expert handling as much as 5 per cent. In the case, therefore, of steam with 10 per cent moisture, the determination of quality might be in error as much as one-half per cent.

Calorimeter Calibrations. For a laboratory calibration exercise three calorimeters of different types are connected by means of exactly the same kind of fittings and valves to the same steam main or receiver. A water-jacket or a device like that shown in **Fig. 44** should be provided to vary the quality of the steam. Tests should be made simultaneously and for the same length of time on the three instruments.

Relative Accuracy of Calorimeters. Because of greater accuracy, comparatively small size, portability, and general indestructibility, throttling calorimeters are generally preferred by engineers. In the order of their relative accuracy steam calorimeters are usually classed as follows: (1) throttling; (2) combined separating and throttling; (3) separating; (4) electric; (5) barrel.

CHAPTER IV

MEASUREMENT OF AREAS

Two methods are in general use for obtaining the area of irregular figures like indicator diagrams: (1) **by measuring ordinates**; and (2) **by means of a planimeter.** Variations of the method of ordinates are known as trapezoidal, Durand's and Simpson's. To find the area by any of these methods divide the figure into an even number of strips by parallel lines. The accuracy is increased as the number of strips is made larger. The notation used in the formulas is illustrated in **Fig. 78**,

FIG. 78. — Diagram of Ordinate Methods.

FIG. 79. — Gramberg's "Line Pattern."

where $\mathbf{y_0}$ is the length of the first ordinate, $\mathbf{y_1}$ of the second, etc., **n** is the number of strips, **w** is the common width of the strips and **A** is the area of the figure. Then the following approximate formulas may be stated:

I. By trapezoidal rule,

$$\mathbf{A} = w\left(\tfrac{1}{2}\, y_0 + y_1 + y_2 + \cdots + y_{n-1} + \tfrac{1}{2}\, y_n\right).$$

II. By Durand's rule,

$$\mathbf{A} = w\,(0.4\, y_0 + 1.1\, y_1 + y_2 + y_3 + \cdots + y_{n-2} + 1.1\, y_{n-1} + 0.4\, y_n).$$

III. By Simpson's rule,

$$\mathbf{A} = \tfrac{1}{3}\, w\,(y_0 + 4\, y_1 + 2\, y_2 + 4\, y_3 + \cdots + 2\, y_{n-2} + 4\, y_{n-1} + y_n).$$

A very convenient method of measuring areas by the use of a "line pattern" has been devised by Granberg.[1] A sheet of tracing cloth or thin celluloid is prepared with parallel lines on it, equally spaced, and with dotted lines (**Fig. 79**) as shown, located at each end of the figure at one-fourth the distance between the unbroken lines first drawn. This "line pattern" is then laid upon the area to be measured so that the

[1] Granberg, *Technische Messungen*, page 48.

ends of the area fall on the " solid " parallel lines at opposite ends. The sum of the lengths of the " solid " lines included by the outline of the area added to one-half the sum of the lengths of the two dotted lines included at the ends when multiplied by the distance between the parallel " solid " lines (**b**) gives the required area.

The various lengths required for both Granberg's and the trapezoidal rules can be conveniently added by laying them off with a dividers one after the other along a straight line and finally measuring the total length of the line.

Areas are also frequently calculated by the **method of mean ordinates,** as given on page 142, for finding the mean effective pressure in engine cylinders.

Planimeters. The most accurate and generally approved method of obtaining the area of irregular figures is by means of integrating instruments called planimeters. Instruments of this kind may differ in many details, yet all of them are based, in theory, on the original Amsler **polar planimeter.**

Polar Planimeters. One of the simplest forms of the polar type of planimeters is shown in **Fig. 80.** It consists essentially of two arms **PO** and **TO** pivoted together at **O.** When in use the point **P** is not to be moved, and is held in place by means of a pin-point upon which a small weight rests. There is a **tracing point** at **T** intended to be moved around the border of the area to be measured. Attached to the arm **TO** is a small graduated wheel **W** carried on a short axis which must be placed accurately parallel to **TO.** Any movement of the arm **TO** except in the direction of its axis will, of course, move the wheel **W** on the paper or other surface on which it is placed in such a way that the amount of its movement gives a record indicating the area measured. A vernier **V** placed opposite the graduations on the wheel, assists in reading the instrument accurately. The arm **TO** is usually made of such a length that the movement of the tracing point **T** around an area of one square inch (for English units) will move the wheel one-tenth of its circumference. Graduations of the vernier indicate usually one one-thousandth of a revolution of the wheel, or in English units one one-hundredth of a square inch.

When the tracing point T is moved around an area in a clockwise direction the wheel will roll in the direction of its graduation, and the area is found by subtracting the final reading from the initial. Amsler planimeters are often constructed with the arm **TO** adjustable in length, so that it can be set to indicate areas in various units, as, for example, square inches, square feet, square centimeters, etc.

The vernier **V** has ten graduations, and the total length of these ten divisions is one-tenth less than the length of those on the wheel so that it represents, counted from zero, so many hundredths of a square inch.

To explain the method of using the vernier, **Fig. 81** has been inserted, showing the wheel **W** and the vernier **V** in a drawing of larger scale than in **Fig. 80.** Readings of the graduations on the wheel **W** are always taken opposite the zero mark on the vernier, so that the reading indicated

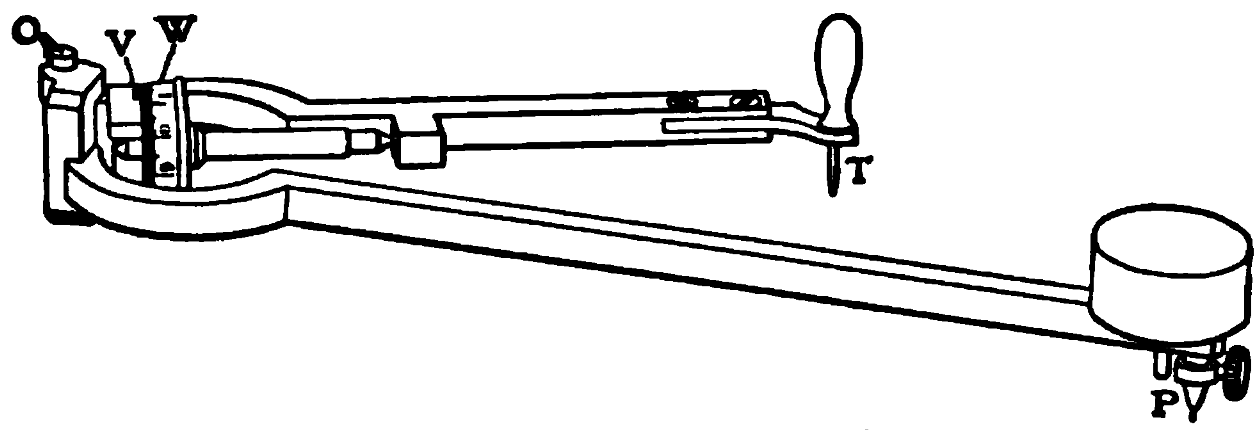


FIG. 80. — Amsler Polar Planimeter.

in **Fig. 81** without the help of the vernier would be a little more than 4.7. The graduation on the vernier which is exactly coincident with a graduation on the roller wheel is the third from zero and indicates three hundredths. The complete reading is therefore 4.73 as determined by the vernier.

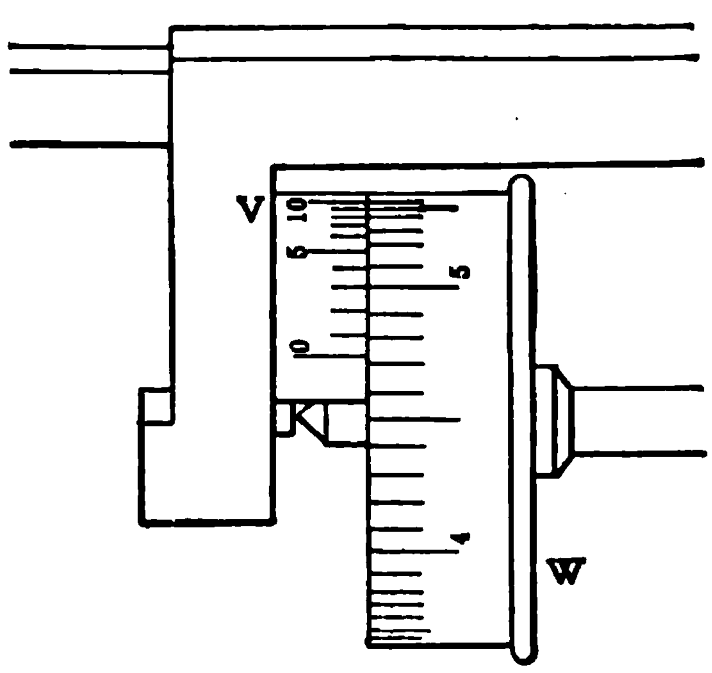


FIG. 81. — Typical Vernier for a Planimeter.

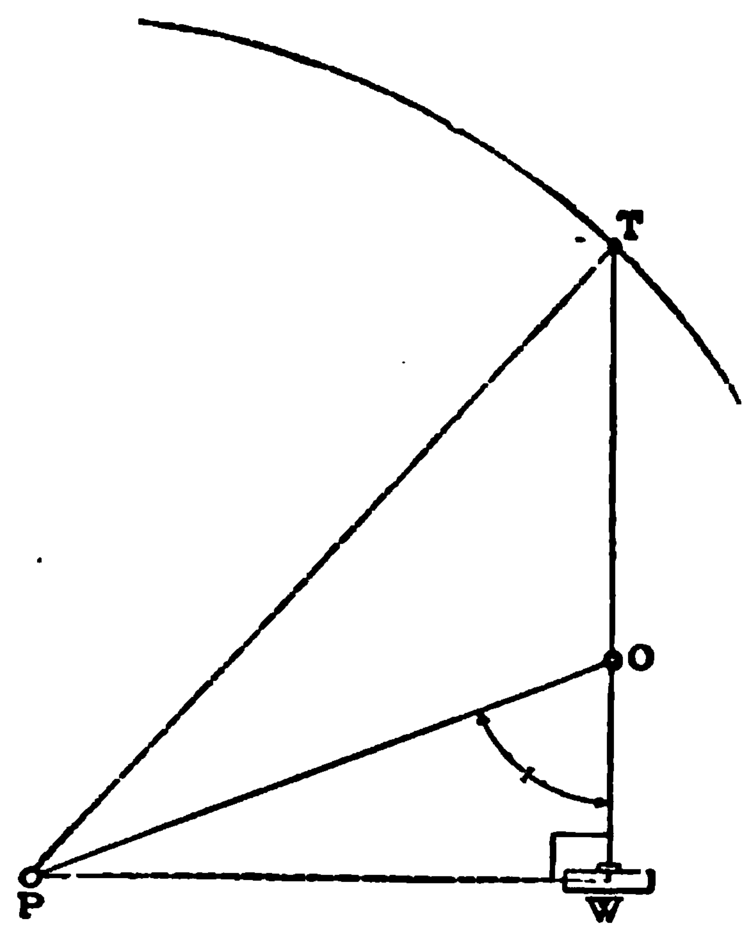


FIG. 82. — Position of the Arms of a Polar Planimeter to Draw the "Zero" Circle.

Theory of Polar Planimeters. As this instrument is constructed neither of the points **T** nor **W** can **pass over** the arm **PO (Fig. 82).** If the arms **PO** and **TO** are clamped so that the plane of the graduated wheel **W** intersects the point **P**,[1] that is, when the angle **TWP** is a right angle, and then the arms thus clamped are revolved around this point, the wheel will be continually slipping without any rolling motion in the

[1] As regards the theory it is immaterial whether **W** is between **O** and **T** or on **TO** extended. Some planimeters are made one way and some the other.

direction of its axis, and consequently it will not revolve. When, however, the arms are not clamped and if the construction of the instrument will permit the tracing-point **T** to be moved out so far that the axis of **W** will lie in the line **PO**, then an arc described by the movement of **T** will produce only a **rolling** motion of the wheel. Obviously with the arms in any position intermediate between that of the clamped right angle and the one with **W** in line with **PO**, the wheel will partly slip and partly roll, the amount of slipping and rolling depending on the size of the angle between the arms. It follows, then, that when circumscribing a closed figure the radial components cause only slipping of the wheel and need not be considered, while the circumferential components produce a resultant rolling which must be taken into consideration.

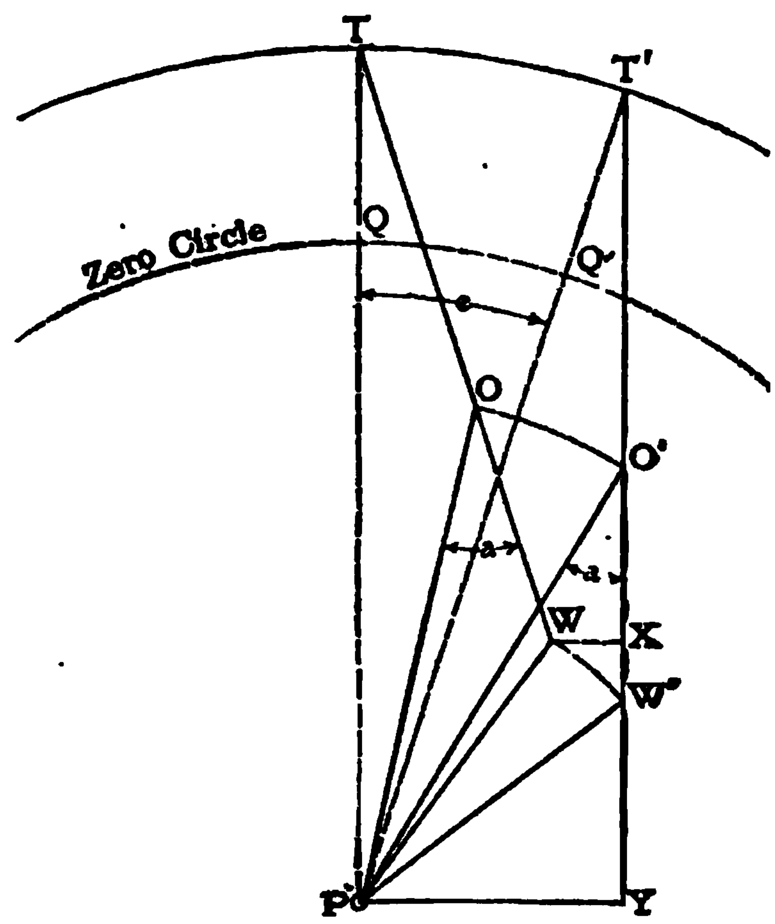


FIG. 83. — Theoretical Diagram for a Polar Planimeter.

The path described by the tracing point **T** when the arms are clamped, as indicated in **Fig. 82**, is called the **zero-circle** for the planimeter. If the tracing-point is moved in any path outside the zero-circle in a clockwise direction a positive record will be indicated on the graduated wheel, while if it is moved in a path in the same direction as before but inside the zero-circle there will be a negative record.

According to the theory of polar planimeters, they are designed so that the rolling of the wheel for a given circumferential motion of the tracing-point **T** is proportional to the area included between the path of **T**, the radial lines from **P** (**Fig. 83**) to the initial and final points of the path taken by **T**, and the arc of the zero-circle included between these radial lines. In other words, the area referred to is **QTT'Q'** in **Fig. 83**. In the discussion of this theory, the circumferential motion of the tracing-point **T** around the point **P**, with the angle **WOP** (marked **a**) remaining always at a constant value, is to be taken up first.[1] Now let us suppose the tracing-point is moved from **T** to **T'** in the figure through a very small angle, **TPT'** (marked **e**), keeping, however, the angle **a** constant; then the graduated wheel **W** will move through the arc **WW'**, partly rolling and partly slipping. The component of this motion producing

[1] In the mathematical discussion following, the graduated wheel will be considered as if it were a part of the arm **TO**, with its plane exactly at right angles to the axis of this arm.

rolling will be perpendicular to the axis of the wheel; or, in other words, this component will be perpendicular to **OT** in all its positions, and without appreciable error for small values it may be represented in **this figure** by the line **WX**, making **WXW′** a right-angled triangle of infinitesimal proportions. When the tracing-point has moved from **T** to **T′** the point **O** has moved through the arc **OO′** and the tracing-point subtends in its movement an angle **WPW′**, which is equal to the angle **TPT′**, marked **e**, which was passed over by **T**. Then the following relation is easily obtained:

$$\mathbf{WW' = PW' \times c.}$$

In this equation the symbol **c** is a constant, expressing the ratio (for a given angle **WPW′**) of the length of an arc to the corresponding radius for any value of this radius. In other words, in terms of the calculus this constant would be expressed in **radians**. In general for every angle there is a constant value which when multiplied by the radius gives the length of the arc for that radius.

The component of **WW′** corresponding to the **rolling** of the wheel is **WX**, which is approximately equal to the arc **WW′** times **cos W′WX.** That is,

$$\mathbf{WX = PW' \times c \times \cos W'WX.} \qquad (8)$$

But if **PY** is drawn perpendicular to **T′W′** produced

$$\mathbf{PW' \cos W'WX = W'Y,} \qquad (9)$$

and combining (8) and (9),

$$\mathbf{W'Y = \frac{WX}{c}.} \qquad (10)$$

Since the angle **WPW′** is very small, **WW′** may be taken as being perpendicular to **W′P.** Now **WX** is perpendicular to **T′Y** and the angle **W′WX** is equal to the angle **PW′Y.** The trigonometric relations reducing the above to terms of the length of one of the arms of the planimeter and the constant angle **a** are as follows:

$$\mathbf{W'Y = \frac{WX}{c} = PW' \cos PW'Y = PO' \cos PO'Y - W'O' = PO' \cos a - W'O',}$$

then
$$\mathbf{WX = c\,(PO' \cos a - W'O').} \qquad (11)$$

This is an expression for the amount of **rolling** of the wheel when the tracing-point moves from **T** to **T′**.

To express the relations required, the area will now be expressed trigonometrically in similar units. From geometry the area of sector **TPT′** = 1/2 **arc TT′ × PT,** but **arc TT′ = PT × c,** or **area TPT′** = 1/2 **c** × $\overline{\mathbf{PT}}^2$.

We can write also,

$$PT = \sqrt{\overline{PO}^2 + OT^2 + 2\,PO \times OT \cos a},$$

$$\text{area } TPT' = 1/2\,c\,(PO^2 + \overline{OT}^2 + 2\,PO \times OT \cos a). \quad . \quad (12)$$

But the area represented by the amount of rolling of the graduated wheel is that part of the sector **outside** the zero-circle (see page 77), and this is the area **TT′Q′Q**. Now the radius r of the zero-circle, referring again to **Fig. 82**,[1] is easily obtained from equations expressing the relations of the sides of the right triangles in that figure for the particular case when there can be no rolling movement. Thus,

$$\overline{PO}^2 = \overline{WO}^2 + \overline{PW}^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

$$\overline{PW}^2 = PT^2 - WT^2 = \overline{PT}^2 - \overline{WO}^2 - 2\,WO \times OT - \overline{OT}^2. \quad . \quad . \quad (14)$$

Combining equations (13) and (14),

$$\overline{PO}^2 = \overline{WO}^2 + \overline{PT}^2 - \overline{WO}^2 - 2\,WO \times OT - \overline{OT}^2.$$

But **PT** = **r**, the radius of zero-circle, therefore,

$$r = \sqrt{\overline{PO}^2 + 2\,WO \times OT + \overline{OT}^2}. \quad . \quad . \quad . \quad . \quad (15)$$

Also from geometry, as explained on the preceding page,

$$\text{Area } QPQ' = 1/2\,r \times c \times r = 1/2\,c \times r^2$$

$$= 1/2\,c\,(\overline{PO}^2 + 2\,WO \times OT + \overline{OT}^2). \quad . \quad . \quad . \quad . \quad (16)$$

Subtracting equation (16) from equation (12),

$$\text{Area } QTT'Q' = c \times OT\,(PO \cos a - WO). \quad . \quad . \quad . \quad (17)$$

Equation (17), which is the expression for the area outside the zero-circle, will be observed to be **equivalent** to the roll of the graduated wheel as given in equation (11), times the length of the arm **OT** from the pivot to the tracing-point. If, therefore, for a given area **A**, we call the reading of the wheel **R** and the length of the arm from pivot to tracing-point **L**, then,

$$A = LR. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

It should be noted further that this equation is independent of any other dimensions of the instrument.

That this demonstration applies to areas not adjacent to the zero-circle or partly inside and out can be readily shown by subtracting in a given case the area between the zero-circle and the required area.

[1] It will be remembered that with the arms of the planimeter in the position shown in **Fig. 82** the tracing-point **T** describes the circumference of the zero-circle.

Area of Zero-Circle by Experiment. The area of the zero-circle of a planimeter may be found readily by passing the tracing-point around the circumference of two circles each larger than the zero-circle. Preferably for this operation the fixed point of the instrument is placed at the center of the circles. If the calculated areas of these circles are respectively A_1 and A_2, and r is the radius of the zero-circle, then since readings of the graduated wheel show only the areas outside the zero-circle represented by R_1 and R_2, we obtain

$$A_1 = \pi r^2 + R_1,$$
$$A_2 = \pi r^2 + R_2,$$
$$2\pi r^2 = A_1 + A_2 - (R_1 + R_2). \quad . \quad . \quad . \quad . \quad . \quad (19)$$

After r has been found[1] it is not difficult to calculate the proper length of the arm **OT** for any linear units (compare equation 15). In fact very many polar planimeters are constructed with the arm **OT** adjustable, so that the instrument can be used for any scale or for various units. The exact lengths required for both the English and metric units (inches and centimeters) are usually stamped on the adjustable arm.

Mean Ordinate of an Area.[2] If we call m the mean ordinate and l the length of a given area A, then

$$A = ml.$$

From equation (18) we have

$$A = LR,$$

whence

$$ml = LR,$$

$$m = \frac{L}{l} R. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

[1] If instead of measuring and calculating the circles both larger than the zero-circle, one of the two is made smaller than the zero-circle, then the reading of the instrument is again the difference between the area of the circle and that of the zero-circle, but the value of this difference is now negative, so that if A_1 is the area of the circle larger than the zero-circle and A_2 is the area of the one smaller, then using the other symbols as before,

$$A_1 = \pi r^2 + R_1,$$
$$A_2 = \pi r^2 - R_2,$$
$$2\pi r^2 = A_1 + A_2 - (R_1 - R_2).$$

Although this latter method does not fall in with the general demonstration so well, it is, however, usually preferred, as it will give greater accuracy than can be obtained with two circles both larger than the zero-circle, unless one of these is made unusually large.

[2] Engineers must calculate **mean ordinates** most often when determining the **mean effective pressure** (m.e.p.) of engine indicator diagrams.

When, therefore, the tracing-point arm is adjustable it may be set as shown in **Fig. 84**[1] to make it equal to the length of the area measured. Then, obviously, the height of the mean ordinate will be equal to the reading of the graduated wheel expressed in the same units. For example, if the subdivisions of the wheel are fortieths of an inch, the result will be the mean ordinate also in fortieths. This scale of the wheel is not determined by the diameter of the portion of the wheel which is graduated, but by the diameter of the **edge** which comes into contact

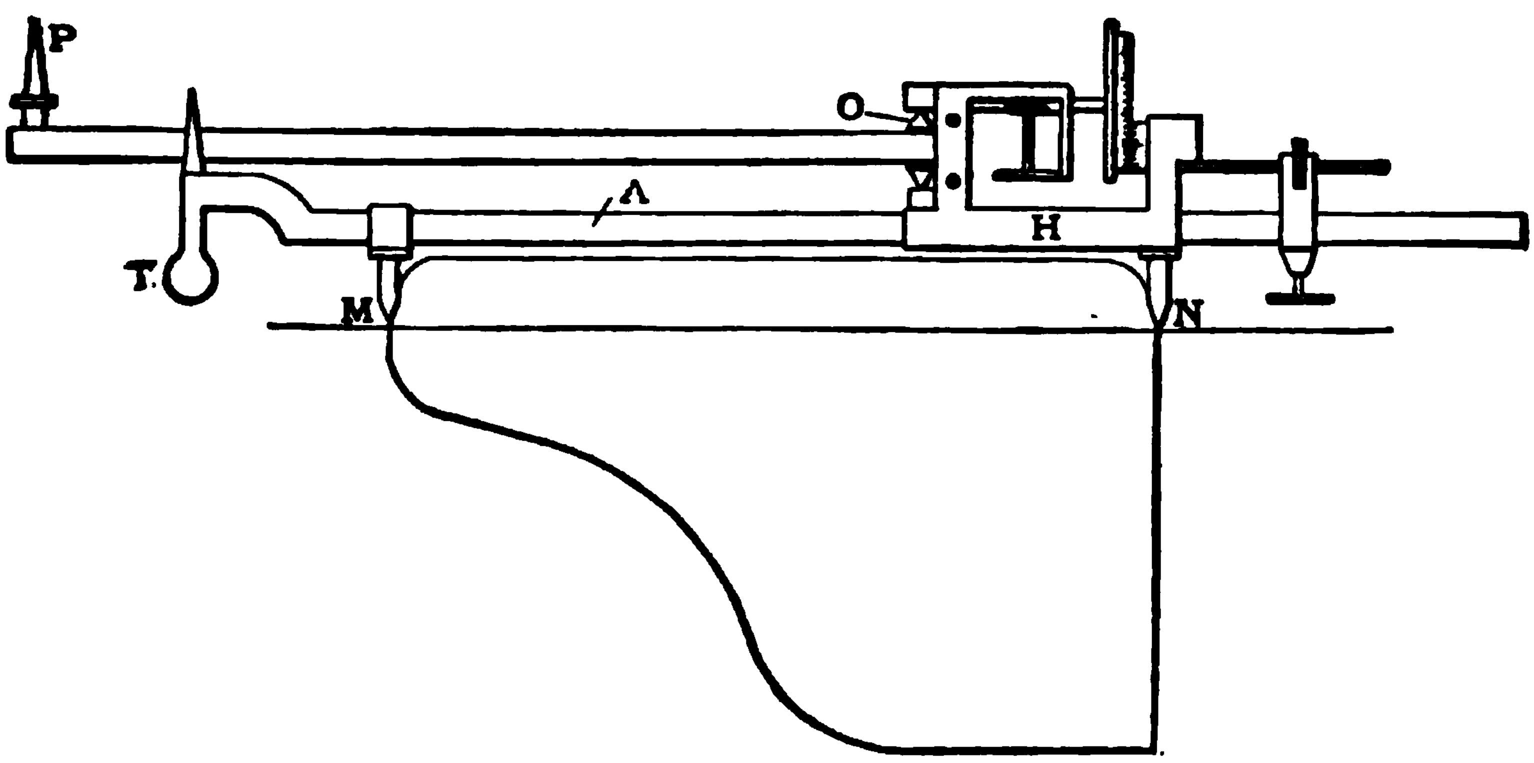


FIG. 84.—Polar Planimeter with Adjustable Arms for the Rapid Determination of Mean Ordinates.

with the surface over which the wheel rolls. If then **d** is the so-called diameter of "rolling" of the wheel, its circumference is π**d.** Now by dividing the number of divisions on the circumference (usually 100) by π**d,** the "scale" of the wheel is obtained. It may also be found by measuring a rectangular area of the same length as that of the **tracer** arm and one inch wide, when the reading from the wheel will give the number of divisions per inch. For those instruments of which the radius of the wheel is one centimeter (.795 inch diameter) and having 100 divisions, the scale is almost exactly 40 divisions to the inch.

Coffin Planimeter and Averaging Instrument. This planimeter is made commonly in two forms, illustrated by **Figs. 85** and **86.** As regards details the former is somewhat the simpler and will be explained first. In principle the two are exactly alike. As will be observed in the figures, this instrument has a single arm to which a suitably graduated

[1] To facilitate the adjustment of the arm to the length of the diagram or area measured, sharp points **M** and **N** are attached to the back of some planimeters. The point **M** is often conveniently placed a short distance away from the tracing-point **T,** and the point **N** must then be the same distance and in the same direction away from the pivot **O.** Then obviously the distance between **M** and **N** will be in all cases equal to the length of the adjustable arm.

wheel is attached on an axis parallel to the line joining the ends of the arm. One of the ends of this arm is for tracing the outline of the area measured while the other slides up and down in a suitable slot. One of the advantages of this instrument over the polar planimeter, although it is not so generally adaptable, is that the wheel is made to move over a specially prepared surface, preventing unnecessary slipping. On mate-

FIG. 85. — Coffin Planimeter.

FIG. 86. — Coffin-Ashcroft Averaging Planimeter.

rials having a rough, fibrous or, worst of all, an uneven surface, the movement of the wheel of any planimeter will not be the same as when rolling over a smooth flat surface.

The Coffin planimeter may be discussed as a special form of the general polar type in which the pivoting point **O**, instead of swinging about the fixed point **P** (**Fig. 80**), moves back and forth in a straight line. The angle between the arms **PO** and **OT**, as indicated by the dotted lines in **Fig. 87**, is really invariable at 90 degrees. Obviously, then, the equation (17) expressing the area traced by a polar planimeter outside the zero-circle becomes, referring to **Fig. 83**,

$$\text{area} = c \times OT\,(-\,WO);$$

likewise equation (11), expressing the roll of the wheel for the Coffin planimeter, becomes equivalent to

$$\text{Roll or record of the wheel} = c\,(-\,W'O') = c\,(-\,WO).$$

Using, as before, in equation (18), the symbols **L** and **R** for, respectively, the length of the arm **OT** and the reading of the wheel, we have, just as for the polar planimeter,

$$A = LR. \qquad (21)$$

As an **averaging instrument** the Coffin planimeter is very much more convenient than the typical forms of polar planimeters. For finding the **mean ordinate** of an area the use of the polar type of these instruments was explained on page 80. The sliding vertical straight edge shown at the right in **Figs. 85** and **86** is for the purpose of making the operation of finding the mean ordinate of an area (or the "mean effective pressure" of an engine indicator diagram) as simple as possible.

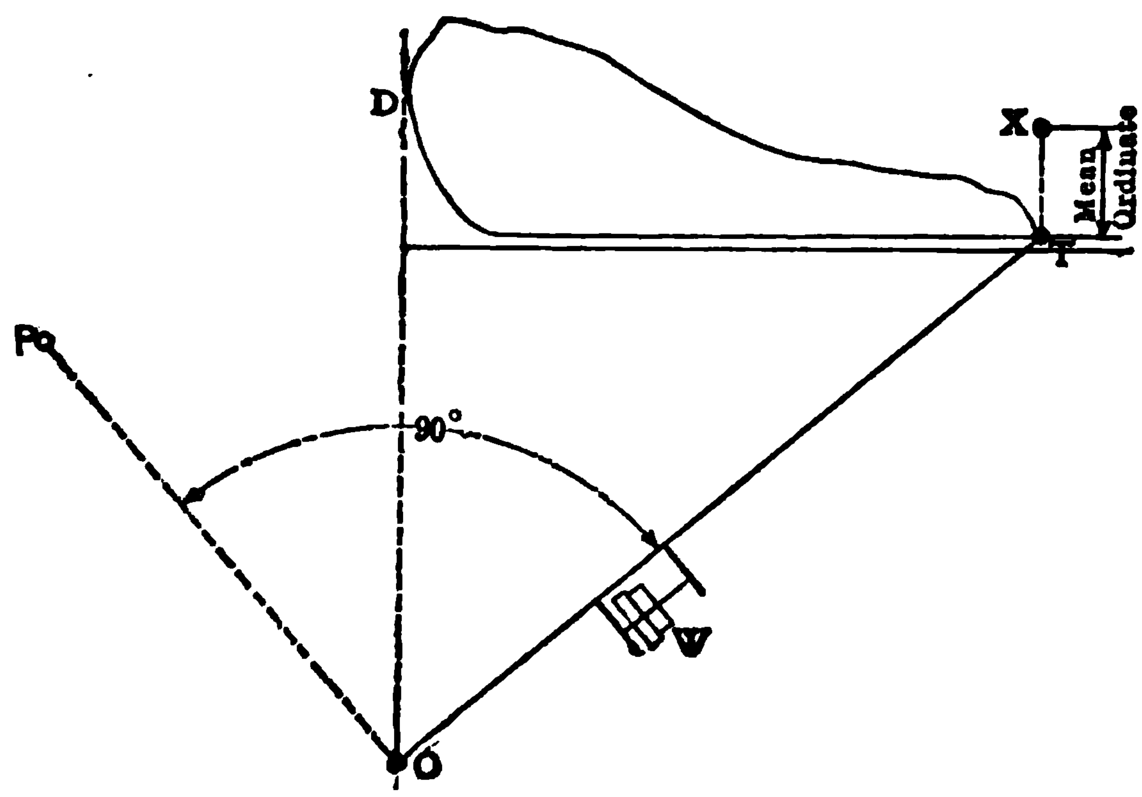


FIG. 87.—Theoretical Diagram for a Coffin Planimeter.

For this operation the straight edges **C** and **K** should be adjusted so that when the tracing pin passes over the extreme end of the area to be measured it will just touch both of them. Now if the tracer is started at either **end** of the area and moved around to the starting point and then moved upward along the vertical straight edge until the reading of the wheel is the same as when starting to trace the area, this last distance traced from the starting point along the vertical straight edge is the **mean ordinate.** To demonstrate this statement the symbols used on page 80 will be continued. Representing the mean ordinate by **m,** the length of the area **A** by **l,** the reading or rolling of the graduated wheel in **going around the area** by **R,** and the length of the arm carrying the tracer by **L,** then as before

$$A = ml.$$

Now, when the tracing-point **T** moves over a vertical line, the angle **DOT,** represented by **Z** in **Fig. 88,** remains constant. If we call the vertical distance moved **V,** and remember that only the movement of the wheel at right angles to its axis produces rolling, then the reading corresponding to the rolling **R** is

$$R = V \sin Z. \qquad (22)$$

But for the position shown in **Fig. 87** when the tracer **T** is at the right-hand end of the outline of the area, we have

$$\sin Z = \frac{l}{L},$$

whence

$$R = \frac{Vl}{L}.$$

Substituting this value of **R** in the general equation (20) for the mean ordinate **m** of a polar planimeter, then,

$$m = \frac{L}{l}\,\frac{Vl}{L} = V. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

This relation can be illustrated more simply, however, by referring to **Fig. 89**, which is a typical indicator diagram from a steam engine. In

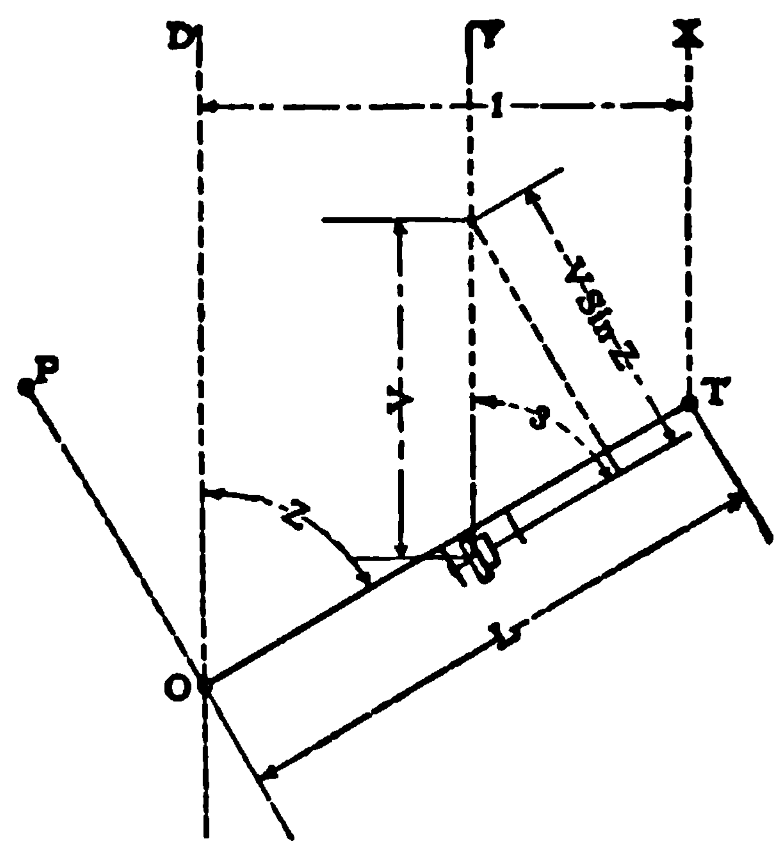


FIG. 88. — Theoretical Diagram for a Coffin Planimeter.

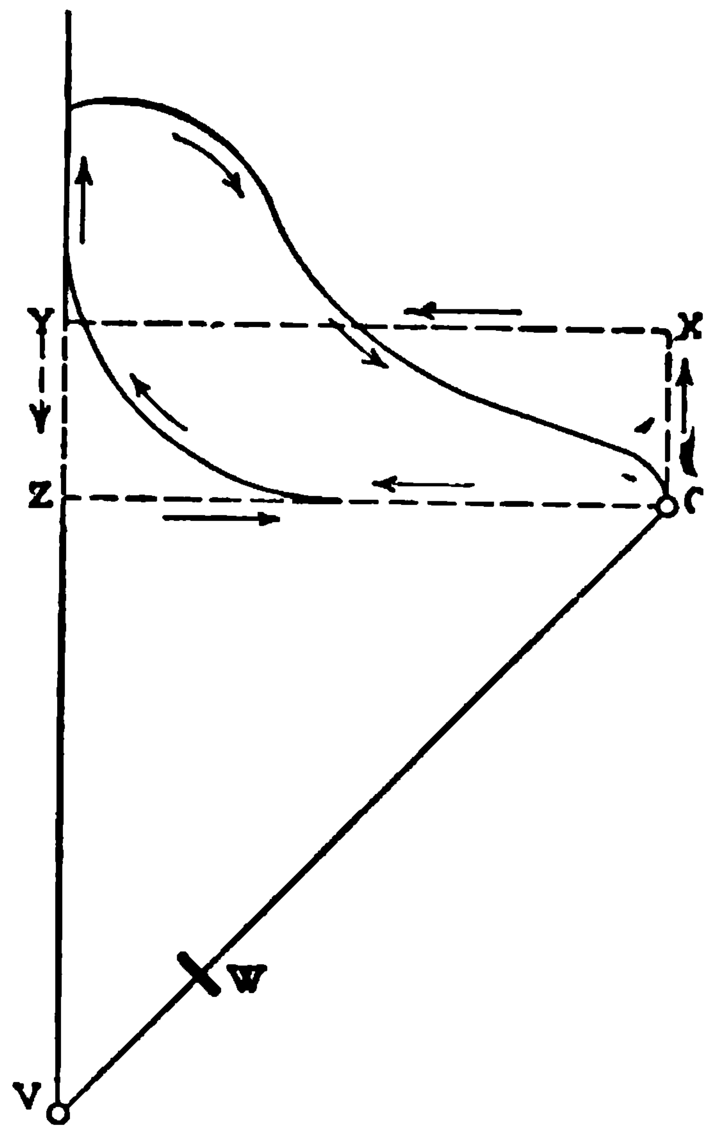


FIG. 89. — Diagram Explaining the Method of Mean Ordinates with a Coffin Planimeter.

this figure the tracing-point of the Coffin instrument is shown at **O**, with the tracing arm represented by **VO**. A rectangle **OXYZ**, indicated by dotted lines, is shown, of which the area is equal to that of the indicator diagram. Starting at **O** and moving the tracing-point around the **indicator diagram** once, the difference in readings is the area. Now if the tracing-point is moved in the opposite direction around the **rectangle** and again back to the starting-point at **O** it will measure a negative area equal to the first area and the reading of the graduated wheel will be the same as when first started around the indicator diagram. The

movement of the graduated wheel as the tracing-point moves from **X** to **Y** is equal and opposite to that in going from **Z** to **O**, so that these two cancel each other. The motion of the tracing-point from **Y** to **Z** requires the axis of the graduated wheel to be parallel to **YV** and consequently during this movement the wheel will not be moved. The only movement that is therefore producing a net change in reading of graduated wheel during the reverse tracing of the rectangle is in going from **O** to **X**. Consequently after going around any irregular area like an indicator diagram in a clockwise direction from the starting-point at **O** at the right-hand end of diagram, if the tracing-point is moved in a vertical direction from the starting-point at **O** until the reading of the graduated wheel is the same as when first started, this vertical distance moved, measured from **O**, will be equal to the mean height of the indicator diagram.

FIG. 90.—A Typical Roller Planimeter.

Although measurements of areas may be made with the Coffin planimeter as with the regular polar types with the area in any position as regards its length and breadth, yet when the mean ordinate is to be obtained, its value in a definite position is required and the area must be placed so that its length with respect to which the mean ordinate is to be obtained will lie along the horizontal straight edge shown in the figures. Then the mean ordinate measured along a vertical straight edge will give the result required.

Roller Planimeters. For the measuring of very large areas a planimeter differing slightly in theory from the polar type has been designed by G. Coradi, of Zurich, Switzerland. It has the advantage of being adaptable for measuring surfaces of indefinite length and as wide as the length of the tracer arm. This instrument is illustrated in **Fig. 90.**

It is supported at three points — the two rollers **R^1** and **R^2** and the tracing pin **f,** or its support **s.** These two rollers are attached to the shaft **A.** On the **face** of one of these rollers is a minutely divided miter-wheel engaging with a small pinion revolving the horizontal shaft carrying the spherical segment **K.** At the center of the frame **B,** and in the same vertical plane with the two shafts already mentioned, a vertical shaft carrying the tracer arm is supported. The spherical segment **K** causes merely by friction contact the movement of the cylindrical "measuring" roller shown at its right. This roller is supported on the auxiliary frame **M,** of which the tracer arm is a part. The "measuring" roller moves back and forth with respect to the spherical segment to correspond with the movement of the tracing-point; but at the same time the rotation of the segment itself imparts rolling motion of the entire instrument.[1]

Calibration of Planimeters. Tests are made by comparing the readings of the instrument with that calculated for a given area. For such calibrations it is necessary to use an area which can be gone over accurately with the tracing-point preferably held mechanically. This is done usually by using a metallic **testing rule,** shown in **Fig. 91.** It is usually

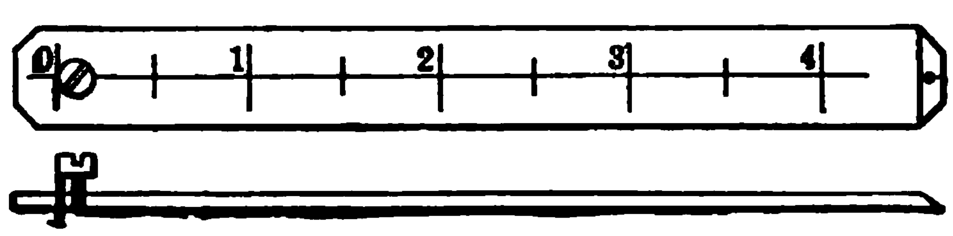


Fig. 91. — Planimeter Testing Rule.

made in the shape of a narrow strip from three to five inches long. At the end marked zero on the graduations a needle point is set which is kept in place by an overlapping screw. At each line of the graduations there is a very small conical hole into which the tracing-point of the planimeter can be placed. The beveled end of the testing rule has the index line set accurately at the starting mark, so that this point can be very carefully located. With the tracing-point **T** in the testing rule and the fixed point **P** of the planimeter in approximately the position shown in **Fig. 92,** observe the reading of the instrument corresponding to the area of the circle described by the tracer moving clockwise, **in the positions shown.**

1st. When the fixed point **P** is on the left-hand side of the tracing point.

2d. When **P** is on the right-hand side.

[1] Since this instrument is not often used by engineers, those interested in its theory are referred to Coradi's book of directions (in English) accompanying each instrument, or to *Handbuch der Vermessungskunde*, by W. Caville.

If the reading obtained is greater in the **first** position than in the **second,** the end of the shaft carrying the graduated wheel nearest the tracing-point must be shifted toward the right to make the instrument accurate, and vice versa. Otherwise the error, if there be one, can be eliminated by taking the mean of the results obtained for the two positions.[1]

Another test to be made, if there is doubt about the accuracy of a planimeter after the axis of the wheel has been adjusted, is to determine whether the settings of the adjustable arm marked on the instrument are correct. For this determination circles with several different diameters can be measured with the testing rule, and if there is a nearly constant **percentage** error, say **x** per cent too large, then the adjustable arm must be lengthened **x** per cent to make the planimeter readings correct, and vice versa.

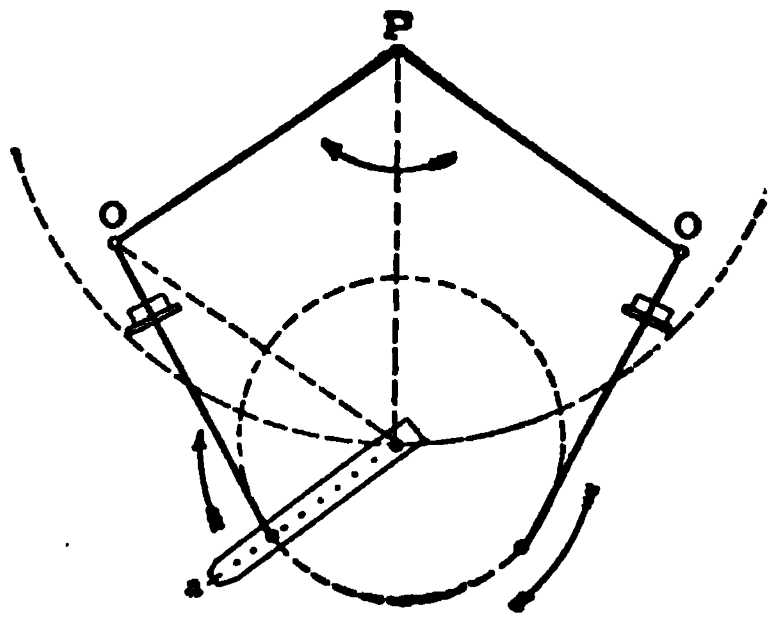


FIG. 92. — Methods of Testing Planimeters.

For accurate results the fixed point P should be placed as indicated by the dotted lines in **Fig. 92,** so that when the tracing-point is near the center of the area to be measured the two arms will be approximately at right angles.

Durand-Bristol Integrating Instrument. This instrument, illustrated in **Fig. 93,** has been recently developed by the Bristol Company for obtaining the average radius of records traced on circular charts of uniform graduations like those used in recording gages, thermometers, etc. It is a simple device for obtaining quickly the average value of pressure, temperature, draft, watts, volts, amperes and other records generally taken on circular charts.

This instrument consists of a wooden base in which there is a metal socket for supporting a rotatable pin slotted for receiving a horizontal shaft to which the integrating wheel is rigidly attached. On this shaft between the integrating wheel and the pin there is an adjustable tracing-point and at the opposite end of the shaft there is a triangular support for the shaft, also adjustable.

The general principle of this instrument is due to Professor W. F.

[1] For ordinary requirements a testing disk can be used in place of the rule, although it is not usually so accurate. On this disk circles of 1, 2, and 2½ inches diameter are usually engraved; and if neither a testing plate nor a disk is available, tests can be made by using circles drawn with a pencil compass on a flat sheet of well-calendered paper.

Durand[1] of Leland Stanford University. Its application hinges on the condition that the chart to be measured has a uniform radial scale, the

FIG. 93. — Bristol-Durand Integrating Instrument for Circular Charts.

same as there must be a uniform vertical scale for indicator and other similar diagrams in order that they can be averaged with the ordinary planimeters. Obviously the mean value of the radius of a circular

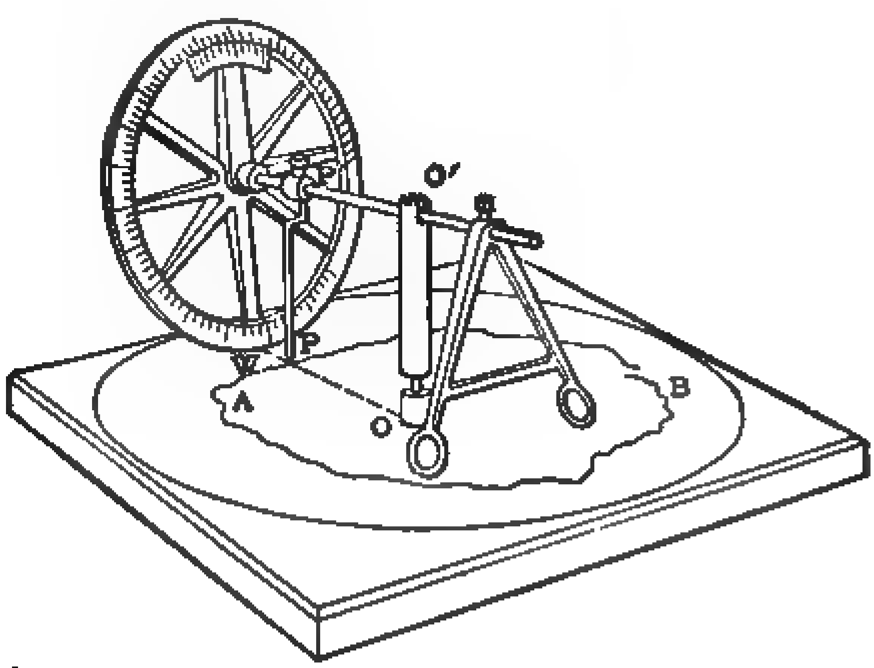


FIG. 94. — Diagrammatic Drawing of Bristol-Durand Integrating Instrument.

diagram cannot be determined with ordinary planimeters, since the area of a diagram in polar co-ordinates is proportional to the square of the

[1] *Transactions American Society of Mechanical Engineers*, vol. 29 (1908).

radius and to the angle.[1] In **Fig. 94 AB** is an irregular curve, considered for this theoretical discussion as traced by a point moving in and out on a **straight radial** line. The center of the chart is at **O**, and at this point there is a socket, in which a rod **O'P** slides freely back and forth, permitting a tracing-point **P** to draw a curve **AB**. A graduated wheel **W** attached to **O'P** serves the same general purpose as the integrating wheel in the ordinary planimeter. Obviously this wheel will be moved only by circumferential motion, and for any radial movement of the rod in the direction of its length it will remain stationary. The amount of movement will be proportional to the radius **WO**, which differs from **PO** by a constant distance **PW**. The resultant movement of the wheel **W** is proportional, therefore, to the angle moved by the arm **O'P** and to the radius **OW** varying from point to point along the curve. Assuming for the present, but as will be shown later, the reading for any part of the curve, as **AB**, to be proportional to the product of the angle subtended between the points **A** and **B**, **AOB**, and the mean radius for the curve between these points, then if this reading is divided by the subtended angle expressed in circular measure the quotient will be proportional to the mean radius. Now if to the value of this mean radius the constant distance **WP** is added, the true value of the radial ordinate **OP** is obtained. When, as is usually the case in practice, the curve **AB** represents values of radial ordinates with reference to a base circle of constant radius as the datum or " zero line," then if the radius of this base circle is subtracted from **OP** the remainder will be the true value of the ordinate. By making **WP** equal to the radius of the base circle, as may readily be done by a suitable adjustment of the instrument, the two corrections will be " balanced " and the mean value of the radial ordinate will be given directly as the quotient of the reading of the wheel and the subtended angle **AOB** expressed in circular measure. For a chart corresponding to twenty-four hours for a circumference, the angular measure to be used as the divisor will be .2618 per hour.

The quantity to be determined in such diagrams is the time-mean of the quantity measured by theradialordinate. But since angular motion is made proportional to time, we may represent the desired mean by the following integral formula:

$$r = \frac{\int r d\theta}{\int d\theta} = \frac{\int r d\theta}{\theta}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

Now, in **Fig. 95**, let **ABCD** denote a curve drawn by a tracing-point

[1] With the ordinary planimeter the mean square of the radial ordinates can be determined, and we can, of course, take the square root of these values, but in most cases this is not the same as the mean radius.

which moves on the arc of a curve shown by **OAV** instead of on a straight radial line. Then let **OV, ON, OM,** etc., denote a series of consecutive positions of the curve **OAV,** at differential angular intervals **dθ**. Then for the actual curved path **ABCD** substitute the broken line path made up of a series of arcs each **r$d\theta$** in length, and the series of differential bits of the curve **OAV** as shown. Then **at the limit** the record of any integrating or averaging instrument will be the same, whether the tracing point is carried along the curve or along the broken line as shown.

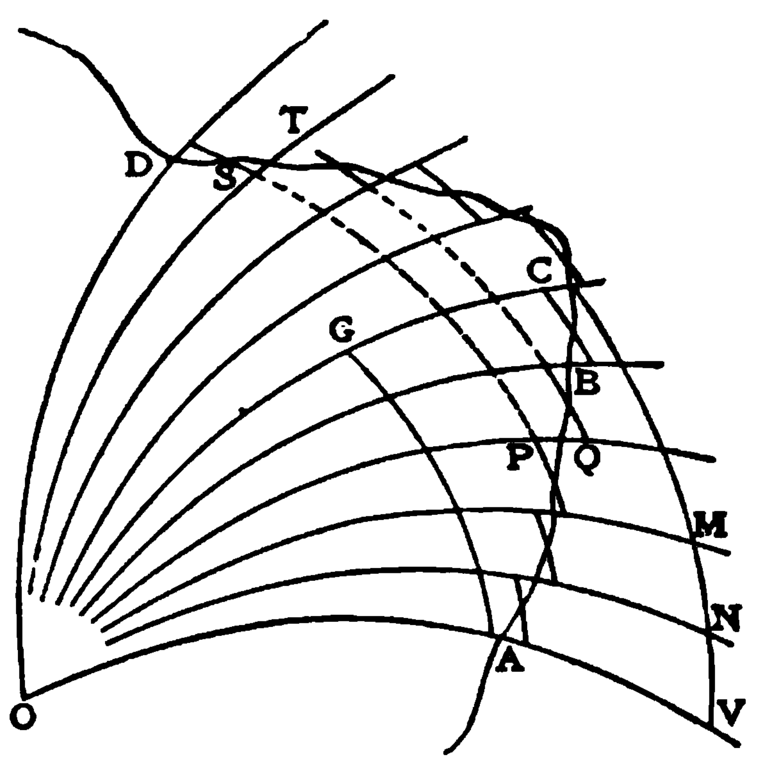


Fig. 95. — Theoretical Curves for Bristol-Durand Instrument.

Then suppose an integrating instrument, as shown in **Figs. 93** and **94,** is applied to such a diagram, and let the tracing-point **P** be carried along the zig-zag path. The record of the wheel will be made up of two parts:

1. That due to the circular arcs **r$d\theta$** and representing by summation the value of $\int \mathbf{r}d\theta$.

2. That due to the differential portions of the arc **OAV.**

Now it is clear that if the diagram extends all the way around from **A** through **BCD** to **A** again the differential elements of the curve **OAV** may be considered as existing in pairs, and that for every element traversed in the **outward** direction there will be an equal element traversed in the **inward** direction. **PQ** and **ST** denote the members of such a pair. The record for such a pair will therefore disappear in the summation; that is, for all the pairs, and also for the diagram as a whole. In such a case, therefore, part "2" above becomes zero and the record of the wheel for the entire diagram consists simply of $\int \mathbf{r}d\theta$.

This reasoning is seen to be entirely general and independent of the character of the path **OAV,** and hence must be true whether it be the arc of a circle, a straight line or any other path.

In case the curve occupies only part of the revolution, as **ABC,** then it is clear that in going from **A** to **C** the record will involve the two parts, "1" and "2" above, and that the latter will remain included in the final result and will represent the summation of the record due to the elements of **OAV** between **A** and **C.** This obviously will be the value of $\int \mathbf{r}d\theta$ for the arc **GC** and it will be canceled by carrying the tracing-point of the instrument back from **C** to **G.** This method of reasoning is **inde-**

pendent of the extent of the arc and is therefore equally true for an entire revolution, even when the diagram does not end at the same radial distance, as at the beginning. In such cases it is necessary only to trace along the arc **OAV** so as to " close " the curve, thus canceling part " 2 " above and finding directly the value of $\int r d\theta$ for a whole revolution.

In all cases the correction for part " 2 " of the record is made by tracing from the terminal point of the curve along the path, representing no change of time to a point lying in a circumference passing through the initial point. This may be stated in other words by saying that to eliminate part " 2 " of the record the tracing-point must start and finish at the same distance from the center, and if the diagram is not of the kind to satisfy this condition then the necessary portion of a path of zero change of time must be used to supplement the diagram. This discussion is independent also of the nature of the curve **OAV**. It may be stated, however, that when **OAV** becomes a straight line the value of the correction becomes zero.

CHAPTER V

ENGINE INDICATORS AND REDUCING MOTIONS

THE engine indicator is simply an instrument showing by graphic diagrams the variations of the pressure in the engine cylinder of steam, gas, air, or whatever the working substance may be. Before James Watt invented the engine indicator (about 1814) he had already used a steam

FIG. 97. — Watt's Original Steam Engine Indicator (Type of 1814).

FIG. 98. — Section of Watt's Indicator.

pressure gage on the cylinder of his engine, and since the movement of the piston in the early steam engines was very slow, he was able to observe with his eyes how the pressure varied during a stroke of the piston. In modern engines the movement of the piston is so rapid, however, that a recording instrument is absolutely necessary.

Watt's indicator is illustrated in **Figs.** 97 and 98. It consists of a cylinder CC (**Fig.** 98) in which the piston P is moved against the resistance of the spring S by steam pressure from the engine cylinder, this pressure being exerted, of course, on the lower side of the piston. A pencil attached to the upper end of the piston rod traces on a sheet of paper a diagram DD, of which the height on any ordinate is proportional to the pressure. The paper is moved back and forth on a slide by a string E moved in conformity with the piston. The instrument was of great service to Watt in perfecting his steam engines. In the modern indicators, of which a few of the best known makes are to be described, there are many improvements over the instrument used by Watt.

Thompson Indicator. Of the engine indicators now in general use the Thompson is the oldest and best known. **Fig.** 99 shows one view of this instrument and **Fig.** 100 shows the corresponding sectional drawing.

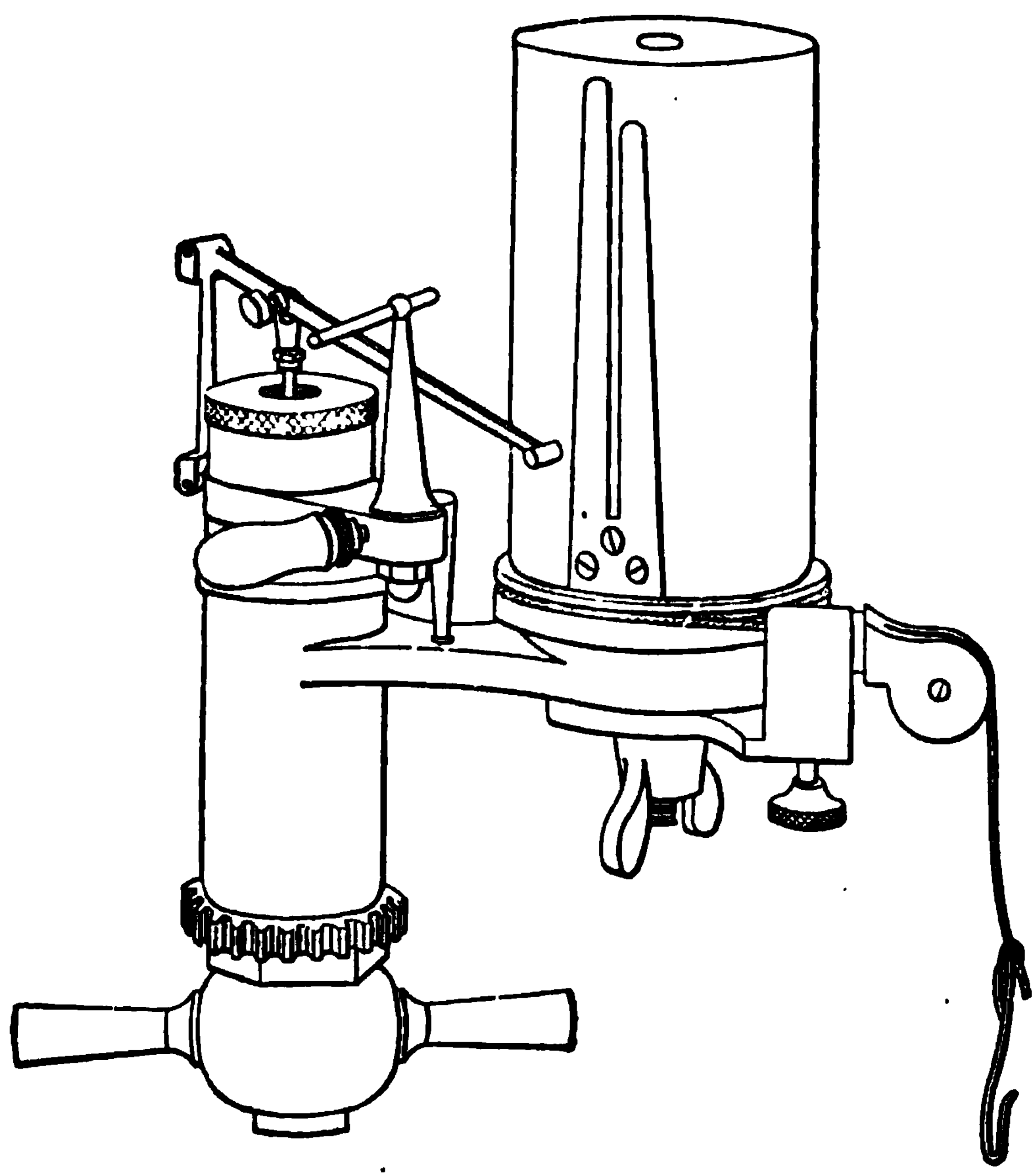


FIG. 99.—Thompson Indicator.

It consists in essential parts of a piston 8 (**Fig.** 100) moving in a cylinder 4. This piston is rigidly connected to the rod 12, which passes up through the cap 2. The motion of the piston rod 12 is transferred to the pencil 23 by means of suitable links designed to make the pencil move parallel to but usually four times as far as the piston 8. The maximum pressure of the pencil on the paper used for the diagram is adjusted by the thread and set-screw on the handle attached to the bracket X.

The method of changing the springs in the various common forms of engine indicators should be well understood by everyone likely to be called on to "indicate" engines. When the work of changing springs is done clumsily or carelessly, a great deal of time is often wasted by the whole party engaged in the test. The method to be followed in changing springs of a Thompson indicator may be stated briefly as follows: The milled-edged cap 2 should first be unscrewed from the top of the cylinder containing the spring and piston. This cap, together with the sleeve and bracket X carrying the pencil lever and linkages, the piston rod, and the piston, can then be lifted from the main body of the indicator. By unscrewing the small milled-headed screw 19 connecting the piston rod with the pencil arm the spring can then be unscrewed, first from the cap 2 and finally from the piston 8. By exactly reversing the operation another spring can be put in the place of the one removed. Changing springs in this instrument is a simple operation. No wrenches or other tools are required. Care should be taken, of course, to screw up the spring firmly against both the cap and the piston. Probably one-half the troubles with indicators in operation arise from loose springs, although not so often, probably, with Thompson indicators as with some other types. The height of the pencil can be adjusted by turning the screw-head 19 up or down on the piston rod.

FIG. 100. — Section of Thompson Indicator.

As a general rule, the spring selected for an indicator should be of such a scale that the largest diagram to be taken will not be more than $1\frac{3}{4}$ inches high; that is, if the maximum pressure will be about 140 pounds, a spring with a scale of 80 pounds per square inch should be selected. Instruction books going with indicators have usually tables showing the spring recommended for a given maximum pressure. Generally a higher card is permissible for light springs and slow engine speeds than for stiff springs and high speeds. The tension of the spring 31 inside the drum carrying the paper for the diagram is varied by loosening

the thumb nut and turning the large milled cap until the proper adjustment is secured.[1]

Crosby Indicators. For high-speed engines and for accurate results the Crosby indicator has long been a favorite with engineers. This indicator is illustrated in **Figs. 101** and **102.** It consists of a piston 8 moving in the cylinder 4, and is connected by means of the piston rod 10 and the link 14 to the pencil lever 16. All of the pencil mechanism (arranged to move the pencil point 23 in a straight line parallel to the

FIG. 101.—Typical Crosby Indicator.

motion of the piston 8) is supported by the links 13 and 15 on the sleeve 3. The indicator spring is fastened at its lower end to the piston by a ball-joint and at its upper end it is screwed into the cap 2. The method of attachment of the springs to the piston by means of the ball-joint is shown more in detail in **Fig. 103.**

In this indicator the spring is changed by first unscrewing the milled

[1] Unless there is a very good reason for a change in the tension of the spring in the drum it should not be altered. Particularly in indicators which have been used a long time the pin holding the spring in place is likely to be much worn, so that if adjusted often the spring may get loose, and then there is usually considerable difficulty in getting it again into its proper position.

cap 2, then this cap, the sleeve 3, the piston rod 10, and the connected parts can be removed from the cylinder 4. By unscrewing the spring by hand from the cap, which, of course, must be prevented from turning, and also from the screw on the swivel head 12, the piston, the spring and the hollow piston rod 10 are detached from the other parts. A socket-wrench of the special form provided in every indicator box of this make is to be slipped over the piston rod to engage with the small nut shown (at 10) in Fig. 102 at the lower end of the piston rod. Then the

Fig. 102.—Section of Crosby Indicator.

piston rod is readily unscrewed from the piston and at the same time the spring is released from its attachment to the piston. Now with the piston rod still in the socket of the wrench, slip the spring to be used over the piston rod until the head of the spring rests in the concave end of the rod. To do this, the wrench must be held upright, and then if the piston is inverted, or, in other words, if it is held so that the end screwing into the piston rod points downward, the piston rod is ready to be screwed into the piston, so that the transverse wire of the spring passing through the bead will be held firmly in the slotted portion of the

socket in the piston. Finally screw the piston rod firmly[1] into place. Before the last operation, the lower piston screw (Fig. 103) should be loosened slightly, and afterward it should be screwed up against the bead just enough to prevent lost motion. It should not be screwed so tightly, however, as to prevent the bead from turning, otherwise the desirable qualities of the ball-joint for securing perfect alignment are lost. Now when the piston rod, spring and piston are again assembled, if the sleeve 3 and the pencil motion attached to it are held in an upright position, the hollow piston rod can be slipped over the threaded portion of the swivel head 11 until the threads on the upper end of the spring engage with those on the cap 2. The spring can then be screwed securely into the cap 2. Then permit the cap to turn in the sleeve 3, and by still turning the spring, screw the piston rod on the swivel head 12 until the top of the rod is nearly flush with the shoulder on the swivel head. The piston and attached spring are now ready to be put into the cylinder by slipping the sleeve 3 into position and screwing down firmly the cap 2.[3]

FIG. 103. — Section of Crosby Indicator Spring and Piston.

The height of the pencil cannot be adjusted to change the position of the atmospheric line without removing the piston from the cylinder of the indicator. It must be done, however, by unscrewing the cap 2 from the cylinder and removing it together with the sleeve 3 and the pencil mechanism. By turning the cap clockwise the swivel-head 11 and consequently also the atmospheric line are lowered. By turning in the opposite direction both are raised. Never try to make adjustments by removing or loosening the pins or screws at the joints 17, 18, 19, 20 and 21. These joints should always be kept tight enough to prevent any lost motion, and occasionally they should be lubricated with refined porpoise oil of the kind usually supplied with indicators.[2]

[1] Special care should be taken when putting a spring into a Crosby indicator that the piston rod is screwed into its socket in the piston P (Fig. 103) as far as it will go; that is, until the extreme upper end of the socket a a is brought firmly against the bottom of the corresponding annular channel b b in the piston rod R.

[2] Persons in charge of tests should always inspect indicators before the steam pressure is put on the springs to observe whether the cap has been screwed down firmly, and whether the pencil mechanism has been adjusted so as to give with a suitable spring a diagram of the proper height.

[3] Inexperienced testers often put the spring and piston into place by merely slipping on the sleeve 3 and without screwing down the cap 2. Then, as a result, when the steam pressure is put on the indicator the piston, spring and pencil mechanism are thrown off with a great deal of force, and some of these parts are sometimes completely demolished.

When using an indicator having the spring inside the cylinder 4 — and this is true

The tension of the spring in the drum is changed very much more conveniently than in most other indicators. For high-speed engines the tension must be considerably greater than that required for those running at low speeds. The tension is adjusted by removing the drum (24) by a straight pull, and turning the knurled nut at the top of the spring (31) after lifting it from its square seat.

Crosby Outside-spring Indicator. Indicator springs arranged outside the cylinder as in Fig. 104 are not subjected to high temperatures and

Fig. 104. — "Tension" Type of Outside-spring Indicator.

are particularly desirable for use with engines operating with superheated steam. There are two principal advantages: (1) The spring can be changed without removing the piston, avoiding an operation often causing confusion and loss of time; (2) the tension of the spring cannot be affected by exposure to very high temperatures. The spring can be

particularly in the Crosby indicator — all adjustments should be made before the steam is turned on the indicator, because the piston, spring and cap soon become very hot, and unless the parts are cooled, preferably by dipping into cold water, they are difficult to handle.

changed when the thumbscrew at the top of the central spindle has been unscrewed.

In **Fig. 105** a slightly different outside-spring arrangement is shown. It is distinguished particularly from **Fig. 104** in having the indicator spring in compression instead of being in tension as in most other outside-spring types. The obvious advantages of the designs having the springs in tension are that springs can be changed much more readily than in other types; and that it is practically impossible if the springs and piston are well made for the spring to buckle over and bind the piston as happens frequently in all types having the spring in compression.

The weakness of the designs having the spring in tension is in requiring a very long and slender piston rod which, being in compression, may have a tendency to buckle over and produce variable errors. As regards temperature effects, one arrangement is about as good as the other.

FIG. 105. — "Compression" Type of Outside Spring Indicator.

Star Brass Indicator — Navy Pattern. The indicator called the "Navy Pattern," manufactured by the Star Brass Co., is shown in **Fig. 106**. In general principles of construction it is like the Crosby indicator illustrated in **Fig. 104**. The most essential difference is in the type of straight-line parallel motion for the pencil lever. It will be observed that this is practically the same as that used in the Thompson indicator (**Fig. 99**).

Tabor Indicator.[1] In the form in which it is now manufactured the Tabor indicator, **Fig. 107**, differs from indicators like the Crosby particularly in the means employed for producing a straight-line parallel motion for the pencil. In this device a roller is attached to the pencil lever and is arranged to move in curved slots on the inside of the rectangular box-shaped part shown in the figure.

As regards the point of flexibility in the mechanism, this is not between the spring and the piston, but, more like the Thompson, is in the ball and socket joint between the piston and the piston rod. Details of this construction are shown in **Fig. 108**.

The principal precaution to observe in the use of this indicator is to

[1] Ashcroft Mfg. Co., Liberty Street, N. Y.

FIG. 106.—Star Brass Indicator—Navy Pattern.

FIG. 107.—Tabor Indicator.

be certain at all times that the roller on the pencil lever moves freely in the curved slots.

Outside spring types of this indicator are also made.

To Change the Spring. The cylinder cap must be first unscrewed, and then this cap, together with the piston, spring, and connected parts can be lifted from the cylinder of the indicator. By removing the small screw under the piston the latter can be unscrewed from the lower end of

Fig. 108. — Section of a Tabor Indicator.

the spring. The other end of the spring can then be unscrewed from the cylinder cap. Another spring is put into the indicator by slipping it over the piston rod with the end stamped **T** uppermost, screwing this end into the cylinder cap and screwing the piston to the lower end. The pencil mechanism must be moved downward until the piston rod enters the piston and the square shoulder enters the corresponding square socket in the piston. In this last operation care must be taken that the rod is firmly and accurately in the hole, and then the screw at the bottom of the piston should be firmly applied.

To Change the Tension of the Spring in the Drum. The drum itself is first removed. Then after loosening the knurled nut on the central shaft and after the drum carriage has been lifted clear of the stops, the carriage can be turned in the required direction to secure the necessary tension and it can then be replaced by lowering into the stops. Care must be taken also that a firm grasp on the drum carriage is not lost, otherwise the spring will become uncoiled and probably also detached.

Fig. 109.—Section of Trill Indicator.

Fig. 109 shows a section of a Trill outside-spring indicator.[1] In principle it is very much the same as the one shown in **Fig. 105.** Important parts are labeled with their proper names, which should be studied.

Bachelder Indicator.[2] **Fig. 110** illustrates an engine indicator which is in many essential parts entirely different from all the types already described. It is so simple in construction that scarcely any description is necessary. The most radical difference is, however, in the form of spring used. This is a flat bar and is arranged with a movable fulcrum which can be adjusted to change the scale of the spring. Although a wide range is obtainable in this way, it has been found unsatisfactory to attempt to use a single spring for all the ranges from the highest pressures to low vacuums. On this account at least two springs, one for high and the other for low pressures, are usually supplied. On account of its heavy parts it is not suitable for high speeds.

Springs are changed by first removing the taper screw shown at the extreme right-hand side in the figure, and then after unscrewing a circular cap on the side of the cylinder the pin connecting the spring to the piston rod can be withdrawn with a small pliers or similar instrument. Usually before the spring can be withdrawn the thumbscrew attached to the movable fulcrum must be loosened. In the ordinary operation of the instrument the piston is not removed.[3]

[1] Trill Indicator Co., Corry, Pa.

[2] Richard Thompson & Co., 126 Liberty Street, New York.

[3] When the spring is calibrated, the piston should be taken out so that a little cylinder oil can be put on it. It is not so necessary for this type when in use on a steam engine, as the oil in the steam will usually provide sufficient lubrication except when the steam is superheated.

The principal difficulty with indicators of this type is that there is always some uncertainty about getting the fulcrum set at exactly the right point. Also if the fulcrum slides easily it may shift during a test. The only safe way is to examine the setting of the fulcrum frequently throughout all tests.

The spring on the drum is conical in form and is adjusted in practically the same way as in the Crosby indicator.

FIG. 110. — Bachelder Indicator.

Precautions for Care of an Indicator. Unless an engine indicator is well taken care of, very soon it will be in a condition in which no reliance can be placed on results obtained with it. That the necessary precautions should be taken is all the more important, because it is one of the most expensive as well as the most delicate instruments used by an engineer in his ordinary practice. The following precautions are particularly important:

1. Before an indicator is used all the working parts, especially the piston, should be carefully cleaned. Then after a spring suitable for the pressure has been attached in its proper position and a little cylinder oil has been smeared in a thin coat on the working surface of the piston, the parts should be replaced. Moving parts of the pencil mechanism should be oiled occasionally with watchmaker's or porpoise oil. It is a very good practice, especially when comparatively new indicators are being used on long tests, to take out the piston of the indicator fre-

quently and smear it with cylinder oil. For lubricating this piston it is a little better to use a comparatively thin cylinder oil of high flash test (like gas-engine oil) than one that is very viscous.

2. Adjust the screw on the handle provided for moving the pencil so that when the pencil is sharp the application of the usual pressure on the handle will give a very fine line.

3. Adjust the length of the indicator cord so that the drum will be neither too loose nor too tight; or in other words, so that the drum will not strike either of the stops when the engine is operating. On a small engine this is most easily tested by observing the diagram when the engine is on each of the dead-centers. If the diagram is either too long or too short the drum will not be moved the required distance, and the indicator diagram will be correspondingly too short and therefore inaccurate. The cord used should be selected with care. It must be of such a quality as not to be stretched appreciably by the forces to which it is subjected. For accurate work and on long-stroke engines fine annealed steel or phosphor-bronze wire, or indicator cord with a wire core,[1] should be used. The length of the cord should be adjusted very carefully and fastened securely so that it will not slip or stretch so as to bring the drum up against one of the stops, making the diagrams too short. This effect can usually be detected by the clicking sound of the drum striking the stop, if there is not too much noise in the room. The experienced engineer, however, will by force of habit invariably put his finger now and then during a test on the top of the drum or on the side of the bracket supporting it to determine whether its operation is satisfactory. Another way to determine a faulty adjustment on the usual crank-shaft type of engine[2] is to measure the lengths of the indicator diagrams. If the cord stretches the diagrams will be variable in length.

4. The atmospheric line should always be taken preferably after the diagram has been made. It is drawn, of course, when the indicator cock is closed. By this order of procedure in tests, the diagram can be more easily taken exactly " on the signal." The length of the diagram must always be measured on the atmospheric line or on a line parallel to it. The indicator cock should be kept closed and the cord to the reducing motion should be unhooked except when a diagram is to be taken. When the cord is unhooked the drum should not be permitted to snap back against the stop. By observing these precautions the useful life of an indicator can be much prolonged.

[1] This indicator cord with a wire core which is guaranteed not to stretch in ordinary use can be obtained from the Athletic Store, State College, Pa. It is the most satisfactory indicator cord obtainable.

[2] In an engine without a crank-shaft like a direct-acting steam pump the length of the stroke and consequently of the indicator diagram is likely to be quite variable.

5. **Immediately after a diagram has been taken it should be removed from the drum and examined.** If there are unusual irregularities in the lines, unaccountable differences in the areas or in the lengths of different cards, the facts should be noted and the best efforts should be made to remedy the faults. Irregularities are usually due to stretching of the indicator cord, grit on the piston, lost motion in the working parts (usually inside the indicator cylinder) or excessive friction caused by overheating of the piston, particularly when used on gas engines.

To correct these faults concerning the piston it must be removed from the cylinder and should then be carefully cleaned and again lubricated with cylinder oil. Before putting the piston and connected parts back into the indicator cylinder it should be observed whether or not all the parts are connected firmly and without lost motion.[1]

6. After a test, the indicator should be removed immediately from the engine, protecting the hands with waste or thick gloves to prevent burns. All the parts, especially those in the cylinder, should be thoroughly cleaned and then put together again without the spring, which should be put away with the other springs in a box provided for the indicator. An indicator should never be handled by taking hold of the drum, as usually it is fastened to the indicator by only a loose slipjoint, and this comes off easily.

7. Before opening the indicator cock to take the first diagram in a test, examine the indicator carefully to see that piston, spring and pencil mechanism are attached securely. A good method is to take hold of the end of the pencil lever near the pencil-point and try to move it up and down. If there is no lost motion observable and the pencil-point seems to be at about the right height for drawing the atmospheric line, it may be assumed that the indicator has been assembled properly. Otherwise it will be observed immediately by this test if the indicator has been put on the engine without inserting a spring, or if the milled nut at the top of the indicator cylinder has not been firmly screwed down. Observe also whether the " union " nut attaching the indicator to the cock is held by at least three or four good threads. Otherwise if this nut slacks

[1] One of the causes of errors in results obtained with indicators not so readily detected is due to the pencil motion not being parallel to that of the piston in the indicator. A simple test for this is to draw an atmospheric line on a card placed on the drum. The card should be at least as wide as the height of the drum. Then after taking out the spring raise the pencil to the full height of the card by pressing lightly on the piston. This operation should be repeated several times at several points along the length of the card. To secure the best accuracy it is desirable to "block" the drum in each position. If the lines drawn are exactly perpendicular to the atmospheric lines there is no error in the pencil mechanism. If the test for perpendicularity is made by a triangle and straight edge, it should be done with the triangle first lying on one side and then on the other, to eliminate any inaccuracy in it. Often the triangles used by engineers are very inaccurate.

back a little the whole indicator may be thrown by the force of the steam pressure against the ceiling of the room. More indicators are worn out and broken by careless assembling than in any other way.

8. One of the best ways to put an indicator card on the drum is to first bend over one of the short edges of the card on a line about a quarter inch from the end and place this end with the line of bending snugly against the top of the longer clip on the drum. Then lap the card around the drum and insert the other end of the card into the upper end of the shorter clip. The card should then be pushed down to the stops in the clips, being careful however to keep it tight and straight, so that there will be no wrinkles. Finally to prevent the card from shifting on the drum, the end of the card under the shorter clip should be bent over carefully and firmly.

SPECIAL TYPES OF ENGINE INDICATORS

Cooley-Hill Continuous Indicator. For many purposes of investigation it is very important to have continuous records showing the variations of the cycles in the operation of an engine. Many devices have been used for this purpose, but as the motion was taken from the crank shaft there was no simple relation between points on these diagrams and the corresponding points in the stroke of the engine. Furthermore, because of the difficult relation, such cards could not be measured with a planimeter. Similar apparatus for the same purpose operated by an electric motor were open to the same objection. To overcome these difficulties a continuous indicator was developed in which the motion was proportional at every instant to the movement of the piston. With a diagram obtained with this instrument it is not difficult to determine the dead-center following release, and the conventional indicator card for an engine is then readily obtained by turning the diagram for the complete cycle back on itself by folding the card or ribbon at this dead-center. If transparent paper is used, the complete diagram can be seen with all the points in their true relative positions as regards the movement of the piston. The indicated horse power can then be readily calculated with the aid of a planimeter. This continuous indicator is illustrated in **Fig. 111**. The indicator cylinder **C**, the piston, and the pencil motion may be of any standard make, as the **collar M**, for attaching the drum mechanism, is adjustable in size so that it can be fitted to indicator cylinders of different diameters. By this arrangement only one drum motion need be provided for using this indicator motion on a number of types of indicators such as would be required for use with steam engines, gas engines, high-pressure air compressors, ammonia compressors, etc. In this apparatus the drum **D** moves forward a given amount with every stroke of the engine. The indicator cord **S** is con-

nected to the indicator reducing motion and is driven by being connected in the usual way to the cross-head of the engine.

The mechanism operating the drum motion is illustrated in **Fig. 111a.** It consists essentially of two miter wheels **B** and **C**, meshing with a similar wheel **E**, to which the pulley **W**, carrying the indicator cord, is attached. At the top of the wheel **B** and at the bottom of **C** are so-called silent

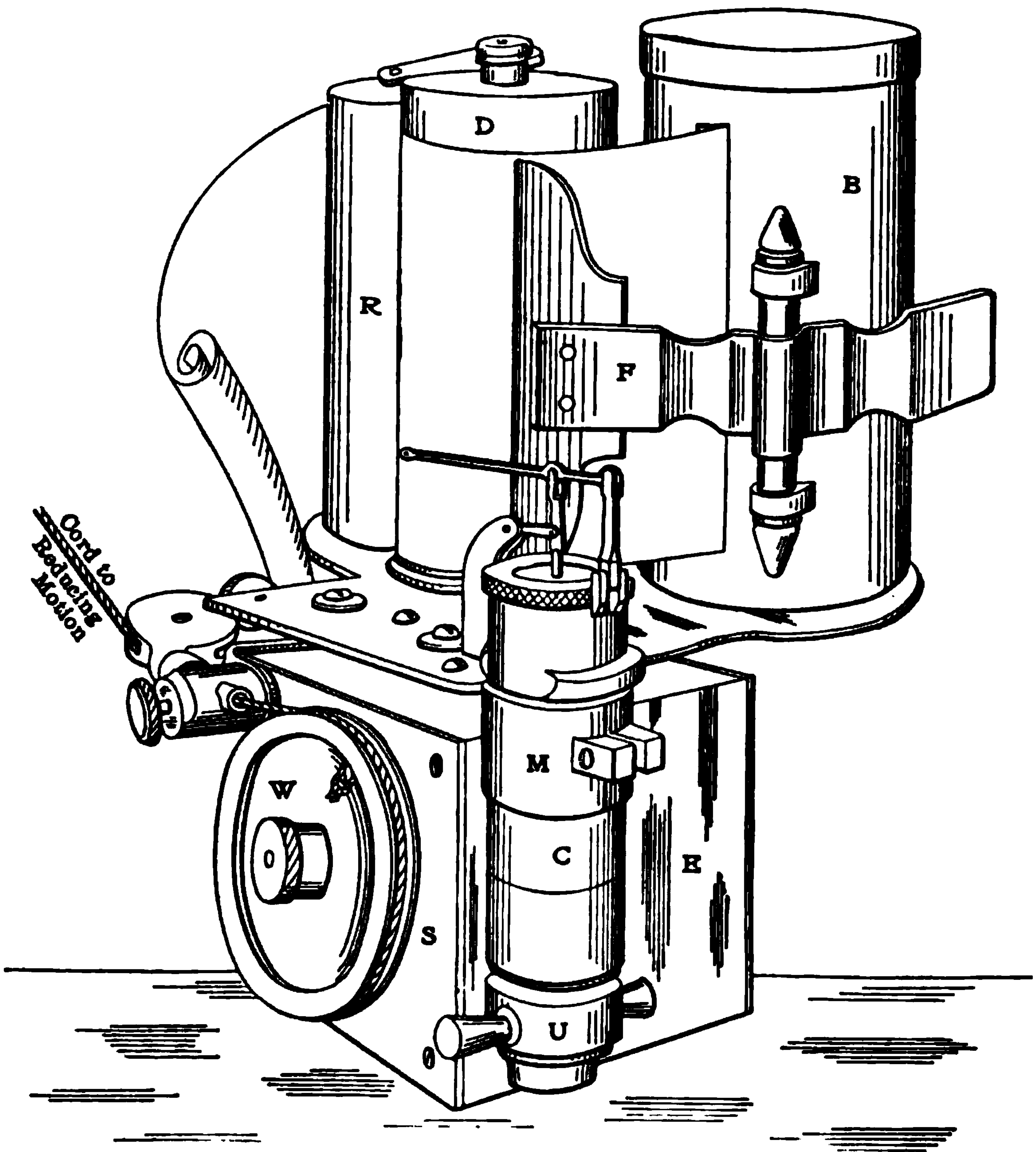


Fig. 111.—Cooley-Hill Continuous Indicator.

ratchet clutches **a, a,** each of which operates in only one direction to grip the collars concentric with the wheels **B** and **C.** Only one of these collars is shown in the figure. Both are rigidly attached to the central spindle **J,** carrying the indicator drum **D** (**Fig. 111**). For example: This central spindle is gripped by the ratchets **a, a** in the wheel **B,** during the "forward" stroke of the engine, and is released during the "backward" stroke. The ratchet in the wheel **C,** on the other hand, grips this same spindle during the "backward" stroke and releases on the "forward" stroke.

In this way the drum **D** is constantly moved on the spindle **J** in the same direction. Neither of the wheels **B** nor **C** is directly attached to the central spindle, and they can move it only when they move in the direction in which they grip their ratchets **a, a,** engaging in the grooves **g, g.**

The miter wheels **B** and **C** are connected to each other by means of a spiral spring enclosed in the casing **D.** This serves the function of the ordinary drum spring in the usual type of indicator for bringing the drum and cord back when the cross-head moves toward the indicator.

Fig. 111a. — Details of Cooley-Hill Indicator.

Optical Indicators. The usual types of indicators operating with a piston are not suitable for engines running at much over 400 revolutions per minute. For higher speeds optical indicators are used. These operate by the deflection of a beam of light from a mirror, the deflection being proportional at any instant to the pressure. When such a device is used on an engine successive indicator diagrams can be readily observed and compared by marking with a pencil the reflection upon a ground-glass plate, and if a photographic sensitive plate is exposed to the beam of light in the place of the ground glass, a permanent impression can be taken, showing at any instant the operation of the engine. Optical indicators are practically the only kind that can be used successfully for indicating the action of modern high-speed **automobile engines.** Every well-equipped automobile testing plant should be provided with one of these instruments. One of the simplest and best apparatus of this kind is illustrated in **Fig. 112.** The indicator is shown in the picture vertically above and connected to the head of the engine. Steam pressure is communicated to the instrument through the usual type of indicator cock supporting it. A system of levers shown (a simple reducing motion) serves for reducing the length of the stroke of the engine to a suitable size for such a small instrument. A glass mirror moved about a vertical axis by the motion transmitted from the cross-head and about a horizontal axis by the pressure in the engine cylinder reflects a beam of light from

a lamp upon a sheet of paper so that the indicator diagram can be traced.

Details of the essential parts of this instrument are shown in **Fig. 112a.** Through the indicator cock the pressure in the engine cylinder is com-

Fig. 112. — Perry's Optical Indicator.

municated to the cored passages marked **A, A.** This pressure tilts the mirror **B,** attached to the thin steel diaphragm **D.** When, therefore, the mirror is still, a ray of reflected light will be seen as a bright **spot** on the screen; but when moved both by the pressure and the motion of the cross-head the conventional indicator diagram is traced. It is very interesting to watch the rapid change of shape of such diagrams as load, speed, pressure, cut-off, etc., are changed. With such an instrument these interesting phenomena in engine operation can be illustrated on a ceiling to a large class of students.

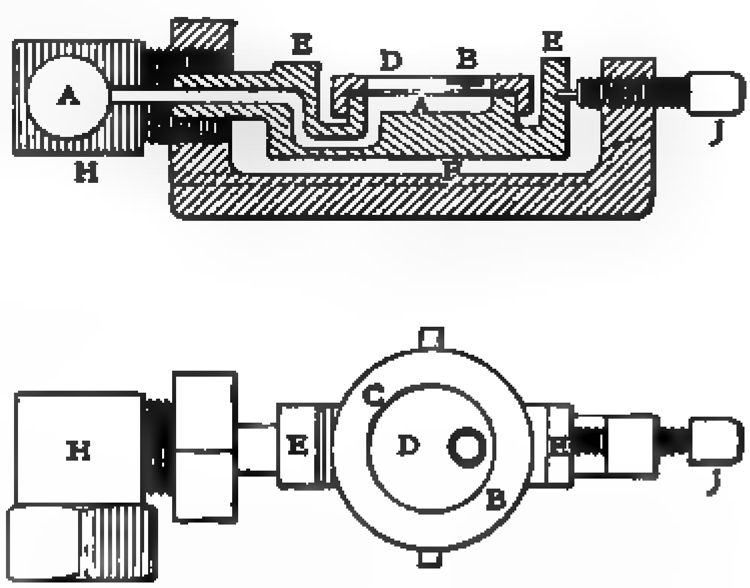


Fig. 112a. — Essential Parts of Perry's Optical Indicator.

Another type of optical indicator intended particularly for high-speed automobile engines is shown in **Fig. 113.** In this instrument the movement of the beam of light is produced by reflection from a small mirror **M** arranged to move in two distinct planes at right angles to each other. In one plane the movement of the piston is accurately reproduced, and in the other plane the movement is proportional to the pressure. Either of these movements or deflections of the mirror, taken alone, would cause the reflected beam of light to trace on the ground-glass plate a straight line; that due to the pressure being arranged to produce a straight vertical line and that due to the motion of the piston a straight horizontal line. But obviously the two movements taken together trace a diagram

FIG. 113.—Section of a "Manograph" Optical Indicator.

indicating at any instant the pressure in the engine cylinder for the corresponding position of the piston. A flexible shaft, attached at one end to the crank shaft of the engine, moves the disk **A** and with it the crank **C**, as well as the small lever **L** attached to it. The free end of this lever is arranged to turn the mirror **M** about a vertical axis by means of the small strut **a**, while the pressure exerted on the diaphragm **D**, as transmitted from the engine cylinder by the pipe **P**, moves the mirror about a horizontal axis by means of the strut **b**. In this apparatus the diaphragm takes the place of the piston and spring in the ordinary type of indicator. These diaphragms, like those used in pressure gages (see page 11), can be made of such thickness that a diagram of satisfactory size can be obtained for high or low pressures. When the diaphragms are carefully calibrated, a reasonable degree of accuracy can be expected. The

relative motions of the mirror in the two planes are set in phase by adjusting the milled screw **S**, operating a small worm wheel serving for changing the angular position of the crank disk **A**, to make the movement of the mirror about the vertical axis correspond with that due to the pressure. There are various methods for determining the proper adjustment for correct "phase relation," but the simplest is to break the ignition circuit on the cylinder to be indicated when the engine is operating. The compression curve will then practically coincide with the expansion line when the adjustment is correct. The principle of operation is shown more clearly in the diagrammatic sketch of **Fig. 114.** Parts are indicated by the same letters as for **Fig. 113.** **Fig. 115** shows the apparatus as it would be set up for indicating an engine. A diagram taken from a gasoline automobile engine is shown in **Fig. 116.**

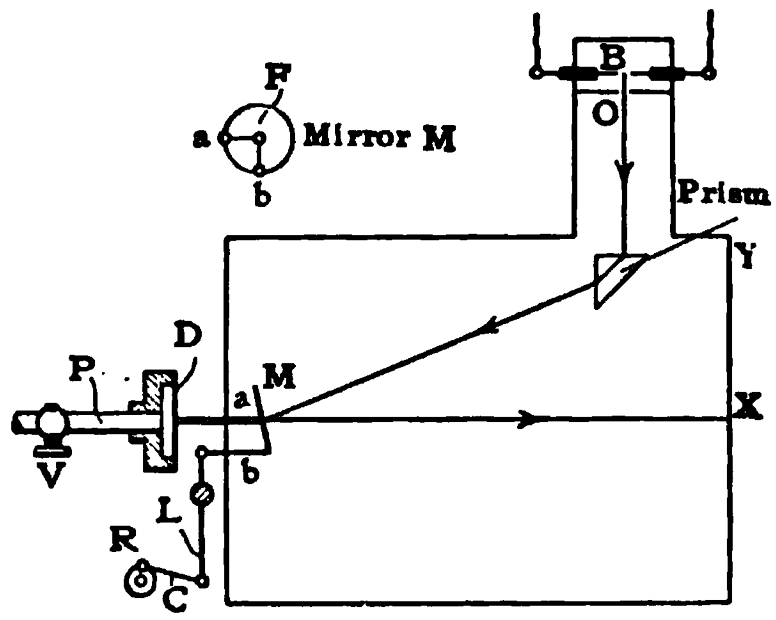


FIG. 114. — Line Diagram of "Manograph."

The Hopkinson optical indicator is shown very clearly in **Fig. 117.** It is essentially similar to the manograph, except that it has a piston **F** instead of a diaphragm and a direct type of reducing motion is used as shown in **Fig. 117a.**

The long tubes connecting the "manograph" type with the engine cylinder are very likely to introduce considerable errors. The time lag between the pressure in the cylinder and that at the diaphragm is very

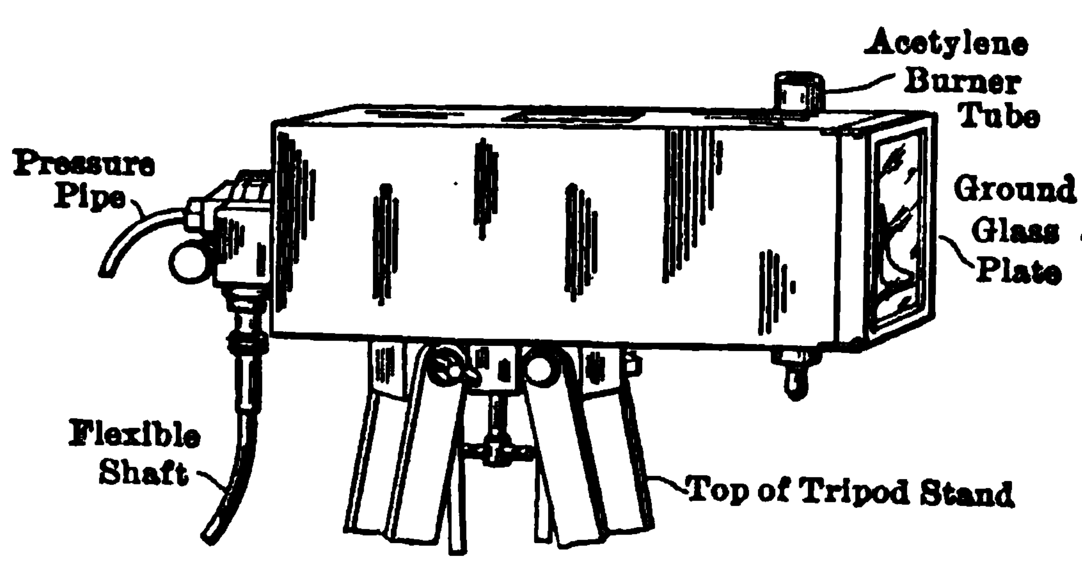


FIG. 115. — "Manograph" Ready for Attachment to Engine.

appreciable at high speed. This error is largely eliminated by adjusting the cyclic relations so that the compression curve observed when the spark is cut off is in phase with the expansion line. But the long tube also throttles very considerably the pressures of the gases and increases the effective clearance of the cylinder. To eliminate these difficulties an ingenious gear device is inserted between the engine shaft

and the small crank **R**. By this means the crank **R** can be retarded behind the engine crank by any phase difference that is necessary. This retardation varies, however, with the speed of the engine and the adjustment

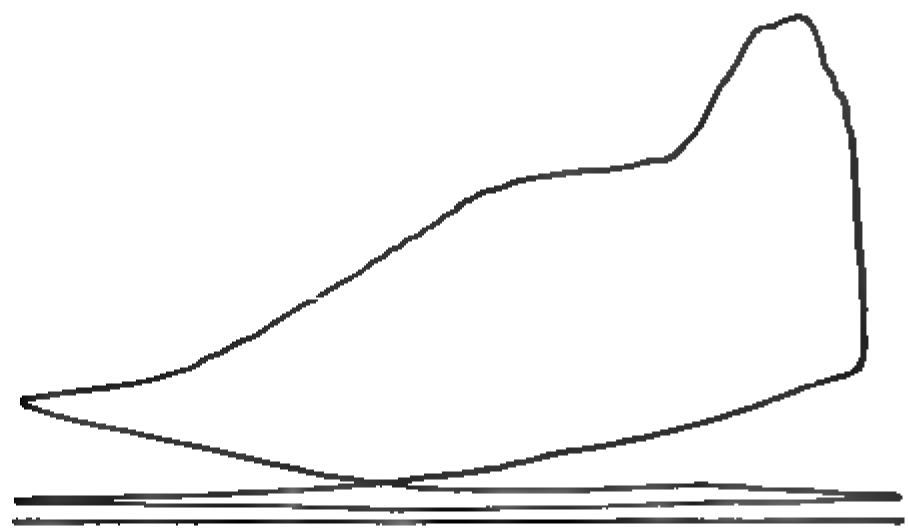


FIG. 116.—Indicator Card taken from a High-speed Automobile Engine with an Optical Indicator.

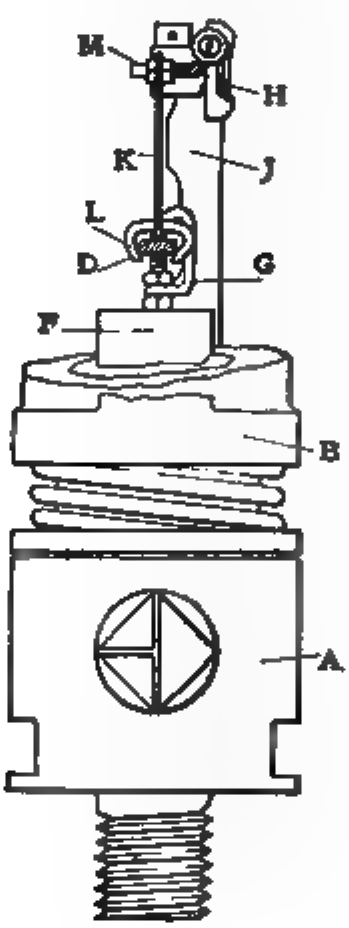


FIG. 117.—Hopkinson's Optical Indicator.

must be made every time the speed is changed in order to get accurate diagrams.

Calibration of Indicator Springs. The pistons of engine indicators are invariably made of a very definite area, usually one-half square inch; and it is possible to calibrate the deflection of the springs with respect

to this area, so that a certain definite pressure per square inch[1] in the cylinder will correspond to a definite deformation of the springs. In English units the pressure on the piston in pounds per square inch corresponding to a movement of the indicator pencil on the diagram of one inch is called the **scale[2] of the spring.** Indicator springs should always be calibrated by the makers. The calibration should be made when they are in the indicator in which they are to be used.

FIG. 117a.—Reducing Motion for Hopkinson's Indicator.

Cooley Apparatus for the Calibration of Indicator Springs. An apparatus similar to the one designed by Professor M. E. Cooley is very generally used for the calibration of indicator springs. One of the latest and more elaborate forms of this instrument is shown in **Fig. 118.** In its essential parts this apparatus consists of a small cylinder **C,** supported on a bracket **B,** a connection **I** at the top of this cylinder for the attachment of the indicator to be tested, and a stuffing-box or gland at the bottom of the cylinder into which a plunger-piston **P** is fitted. The lower end of this plunger rests on a sensitive platform scales. Any pressure in the cylinder **C** can therefore be weighed. Steam may be admitted to the cylinder through a pipe **E,** and is exhausted through the pipe **A.** By adjusting the globe valves on the pipes **A** and **E** any pressure desired can be secured in the cylinder **C,** and this same pressure is, of course, exerted both on the piston of the indicator above and on the plunger **P** below. This plunger is usually made with an area of one-half square inch. For a plunger of this area, then, if for a given pressure the scales balance at 10 pounds, the pressure in the cylinder **C,** and on the piston of the indicator, is 20 pounds per square inch. To eliminate friction[3]

[1] Indicators are always designed to relieve the pressure above the piston due to leakage around it, so that on this side there is always atmospheric pressure.

[2] Instead of "scale" the word "number" is often used. That is, a spring of which the scale is 40 pounds would be called "No. 40."

[3] The operation of the apparatus is usually much improved by pouring, just before the steam valves are opened, a few drops of cylinder oil into the cylinder **C** through **I** to lubricate the plunger.

as much as possible, the plunger **P** should be kept spinning when observations are being taken. For this purpose a hand wheel **K** with considerable mass, for its " fly-wheel " effect, is provided on the shaft of the plunger. A more uniform motion of the plunger is obtained, however, by having the hand wheel grooved to take a small belt to be driven by an electric motor **M**. The plunger is supported usually on a ball-bearing joint set in a low pedestal **L**.

By connecting a pipe **E** to a suitable manifold or similar fitting, to which are attached three separate pipes supplying respectively steam, air, and water under pressure, an indicator can be tested with varying

FIG. 118. — Apparatus for Calibrating Indicator Springs.

pressures under the actual conditions in service; that is, when used for steam, air or water. American engineers always prefer making calibrations of indicators under conditions as nearly as possible those pertaining to their ordinary use. The Power Test Committee of the A.S.M.E. recommends also the following procedure:

To bring the conditions approximately at least to those of the working indicator, the steam should be admitted to the indicator in as short a time as practicable for each of the pressures tried, and then the indicator cock should be closed and the steam exhausted before another pressure is tried. By this means the parts are heated and cooled as under working conditions. For each required pressure open and close the indicator cock a number of times in quick succession, then quickly draw the line for the

desired record, observing at the same instant the "reading" of the standard used for comparison. A corresponding atmospheric line is to be taken immediately after each pressure line.

Indicator springs for gas and oil engines should also be calibrated with the indicator in as nearly the same condition as to temperature as exists when it is in use. A simple way of heating recommended is to subject it to steam pressure just before calibration. Compressed air is a suitable fluid for the actual calibration, being preferred to steam as it brings the conditions as nearly as possible to those of practice when the indicator is in actual use in gas or oil engines.

In Europe an apparatus like Fig. 119 is used a great deal. It is essentially the same as the dead-weight gage testers described on page 17, except that there is a connection for an indicator. Pressure is applied by loading weights on the platform P resting on the plunger. In this apparatus the gage G serves merely as a means of checking and avoiding mistakes. American engineers object to this method because the calibration is made when the indicator is under nothing like the conditions of service, at least as regards temperature. Many engineers calibrate their indicators by comparing them with a good test-gage which has been carefully calibrated with a dead-weight tester. The gage and indicator are put on the same pipe carrying high-pressure steam. The movement of the pencil of the indicator is carefully observed, and compared with the reading of the gage. This latter is the method suggested by the Power Test Committee of the A.S.M.E. in their report in Nov., 1912.

Indicator G

FIG. 119. — Dead-weight Tester for Indicator Springs.

A simpler form of the Cooley apparatus intended for the so-called "dry method" of testing is shown in Fig. 120. A suitable fitting for receiving the indicator I is supported on the bracket B. The legs of this bracket span over a sensitive platform scales S. A small rod R rests at its lower end on a small pedestal standing on the platform of the scales S. On the top of this rod there is a cap supported on a small conical bearing to give some flexibility. This cap is made to fit easily into the lower side of the piston in the indicator. The indicator itself is attached to the top of the hand wheel W. Then when the hand wheel is screwed downward the indicator comes down with it and compresses the indicator spring. At the same time a pressure is exerted on the rod R which can be balanced on the scale beam. When a force is applied to compress the spring in the indicator, the magnitude of the force can be determined by weighing the pressure on the scales. If the area of the piston in the indicator is one-half square inch, then twice the weight on the scales is the pressure exerted in pounds per square

inch. Heat can be applied to the indicator by passing steam through a rubber or flexible copper tube wrapped around the cylinder.[1]

Method for Calibration of Springs. After cleaning the internal parts of the indicator, inserting the spring to be calibrated, and oiling the piston with cylinder oil, the indicator is to be attached to the indicator cock on the calibrating apparatus. Before putting the card on the in-

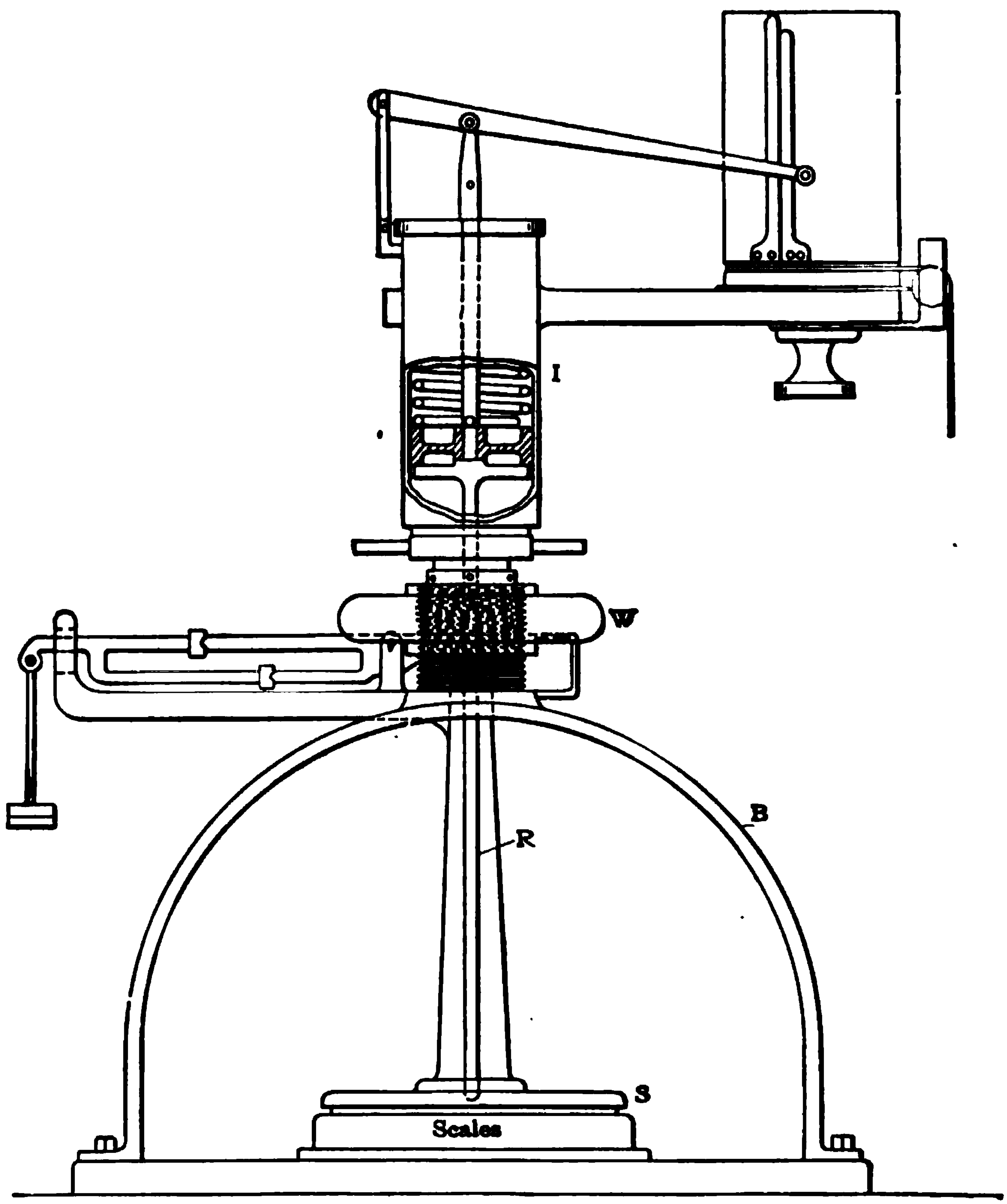


FIG. 120.—Apparatus for "Dry" Method of Indicator Testing.

[1] Some engineers, particularly in Germany, advocate that an indicator should be slightly jarred just before each calibration line is drawn, intending that this jarring is equivalent to the vibrations which an indicator receives when in service on an engine. Since, however, readings are taken with both increasing and decreasing pressures it is doubtful whether this additional work is necessary on an apparatus like Fig. 118. If on the other hand a dead-weight tester like Fig. 119 or the method of comparison with a test gage (page 115) is used, tapping both the indicator and the gage is probably very advisable.

dicator drum on which the record is to be made, two approximately parallel and vertical lines should be drawn on it about one-half inch apart, similar to the lines **AB** and **CD** in **Fig. 121.** Meanwhile the indicator should be thoroughly warmed if a calibration with steam pressure is to be made. Then with the indicator cock and the valve on the steam pipe **E** (**Fig. 118**) closed and the exhaust pipe **A** open, draw the first calibration line on the card. This should be made by setting the pencil point at **D**, and then by pulling the cord attached to the drum draw a line **crossing** the vertical line **AB.** With springs of which the scale is 40 pounds or less, a similar record should be made for increments of every 5 pounds per square inch change in pressure, while for higher scales the increments may be made 10 pounds. If with in-

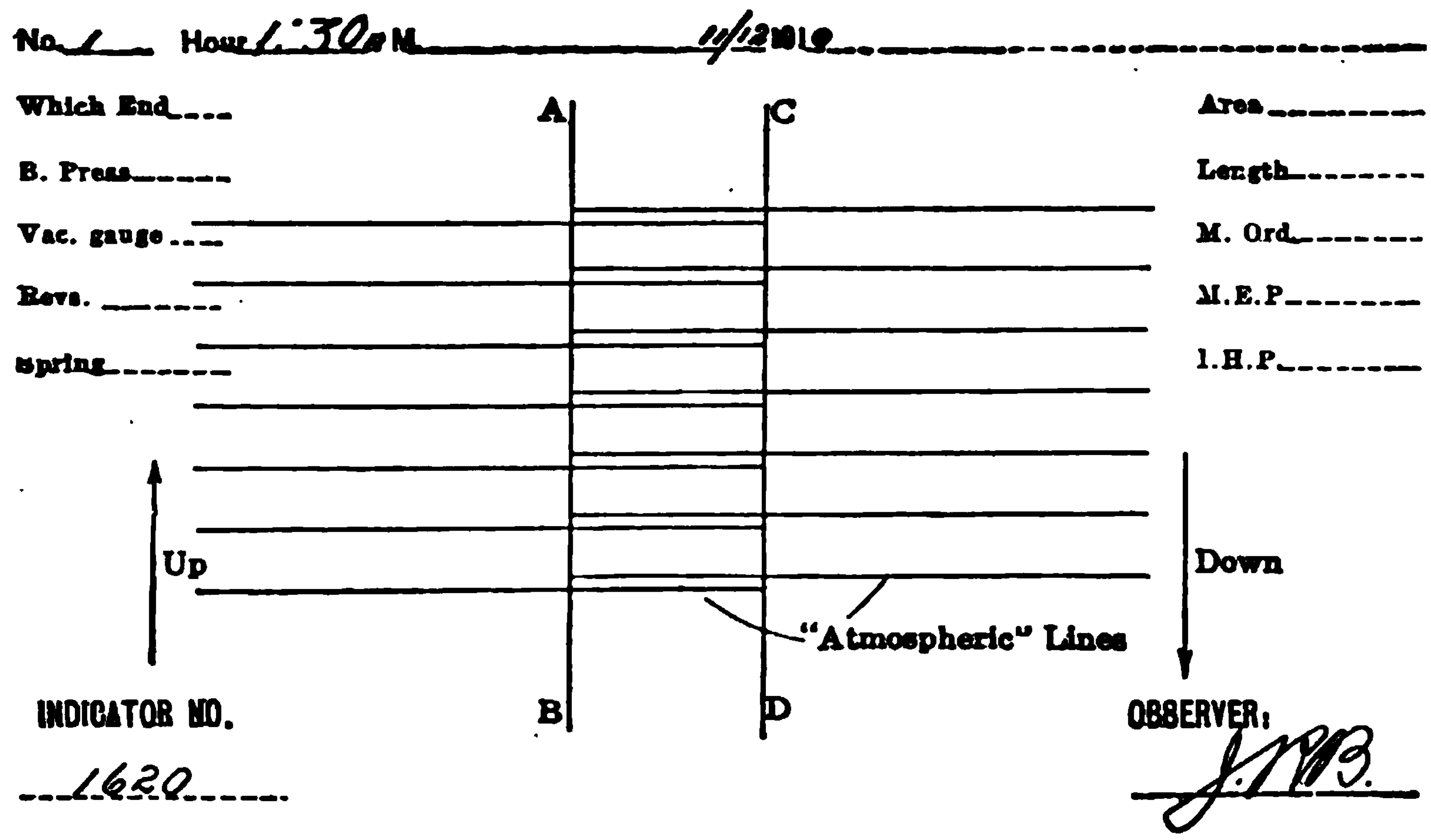


Fig. 121.—Sample Card Illustrating a Test of an Indicator Spring.

creasing pressures the lines are drawn toward the left, then with decreasing pressures they should be drawn toward the right with equal increments, beginning at the opposite vertical line **AB.** By this method the corresponding lines for equal pressures will be immediately over each other between the two verticals. With an accurate scale, graduated preferably to one one-hundredths inch, measure between **AB** and **CD** the distance from the atmospheric line first drawn to the various "pressure lines," and record the results. Care should be taken that with the increasing increments the pencil **rises** to the required pressure and that with decreasing increments it **falls** to these pressures. In other words, if when the lines for increasing pressures are being drawn, the pressure rises too rapidly to draw the line at the proper time when the scale beam is just balancing, then the pressure should be again reduced below the value required, so that the pencil will be again ascend-

ing when the line is drawn. Similarly for decreasing pressures, if the pressure gets too low, it must be increased and again brought down to the required value.[1] If the results obtained do not seem to be consistent, the difficulty is probably due to passing the required pressure so rapidly that the lines have not all been drawn at the proper time.

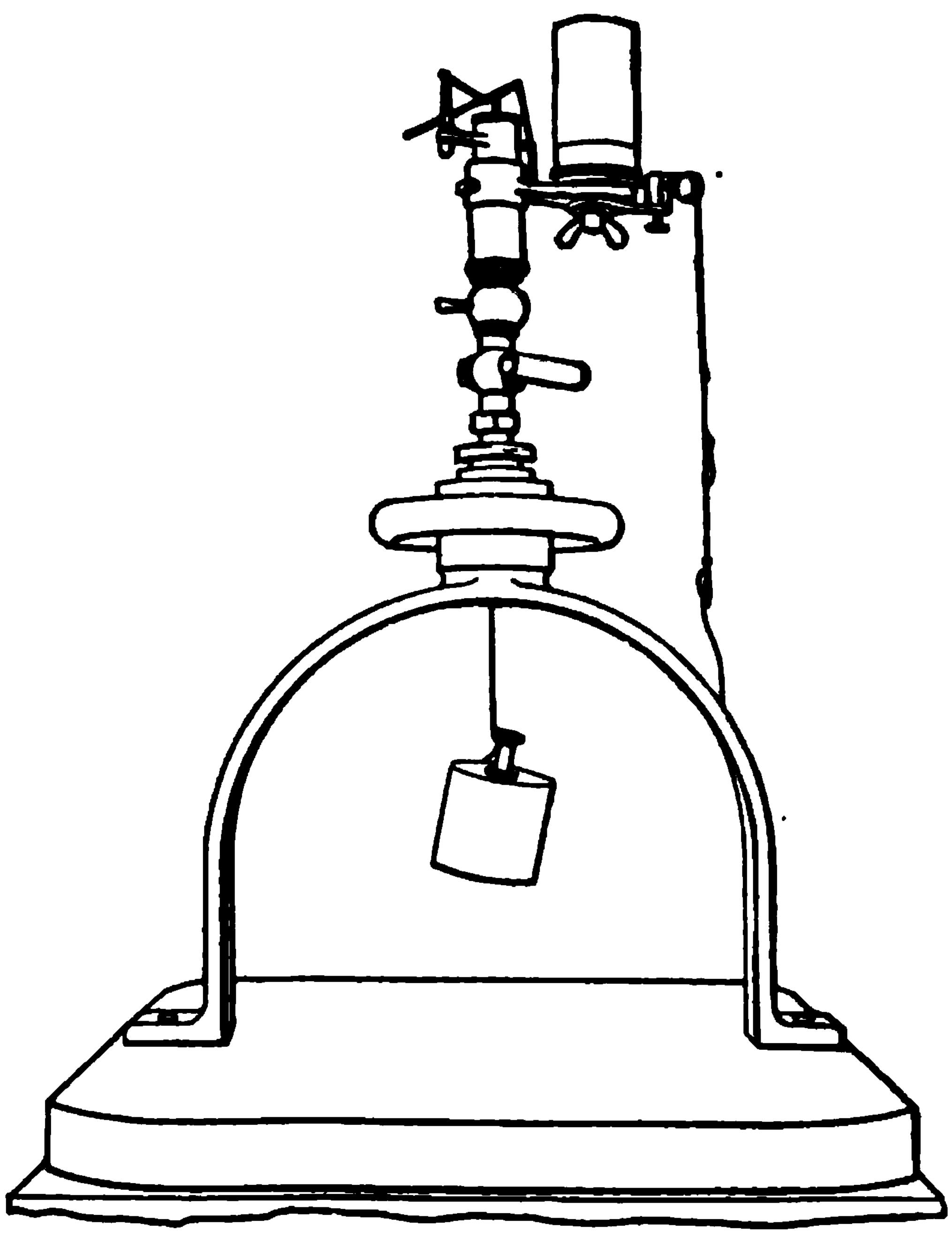


Fig. 122. — Apparatus for Testing Indicator Springs in Tension.

The difference between the lines for increasing and decreasing pressures shows the amount of friction and lost motion in the indicator.[2] The error of the instrument is obtained by comparing the mean ordinates of the card thus obtained with the actual pressures as determined by weighing. From time to time the accuracy of the platform scales should be determined by testing with standard weights. For dependable results two calibrations, each "up and down," should be made for each spring and the results compared.

When indicators are used for pressures which are never less than atmospheric, the springs are in compression and apparatus of the form described are satisfactory; but when indicators are used on the low-pressure cylinders of engines, the springs are usually in **tension.** For this service a slightly different device must be used for calibration. A suitable apparatus is shown in **Fig. 122.**

The indicator **I** is supported on a bracket similar to the one used in the apparatus shown in **Fig. 120.** A short steel rod (about No. 18, B. & S. gage) is attached to the lower side of the piston in the indicator by screwing into a hole tapped centrally. Now if weights are suspended

[1] The same precaution must be observed in beginning the test. To be sure that the pencil and piston have not been falling instead of rising, the piston rod should be pushed down lightly before the atmospheric line is drawn.

[2] Half of this difference, to be more accurate, represents the friction and lost motion in any one position.

from the end of this rod[1] the spring can be calibrated in tension by drawing lines on a paper card placed on the drum in the same general way as when the spring was tested in compression.

A suggested form for the arrangement of data for these calibrations is given below.

CALIBRATION OF INDICATOR SPRING (COMPRESSION)

In Indicator No........................

Rated scale of spring..
Diameter of piston of testing apparatus..ins.
Area of piston of testing apparatus...sq. ins.
Identification marks on spring...
Observers { .... Date....................
{ ....

No. of Reading.	Net Weight on Scales, Lbs.	Actual Pressure on Piston, lbs./sq. in.	Ordinates or "Heights" Measured on Card, Inches.			True Scale of Spring,[1] (3)+(6).	Remarks.
			Up.	Down.	Average.		
1	2	3	4	5	6		

[1] "The calibration of a spring should be made for at least five equidistant points. For ordinary work the arithmetical mean of the various results should be taken for the average scale." — Report of Power Test Committee of A.S.M.E.

Curves. Results should be shown graphically for calibrations of indicator springs by plotting for abscissas the actual pressures in pounds per square inch and for ordinates the corresponding average height, inches, above the atmospheric line. Use very large scales as otherwise these curves are of little value.

Calibration of Indicator Springs with the Mercury Column. The method to be followed in calibrating indicator springs with a mercury column is essentially the same as described on page 19 for the calibration of pressure gages. After the indicator has been cleaned and oiled, it should be attached to the testing drum or cylinder by means of an indicator cock. Then while the indicator is being heated to the temperature of the fluid medium used (steam, air or water), the paper

[1] The weight of this rod and wires or strings supporting the weights must be added to them to get the correct tension. If, however, only the true scale of the spring is desired, as is usually the case, the weight of these parts need not be considered, provided, of course, the atmospheric as well as the other lines are all drawn with these parts attached to the piston.

card can be put on the drum after first drawing two vertical lines one-half inch apart, as explained when describing the apparatus shown in **Fig. 118.** Following these same instructions the atmospheric and other pressure lines are drawn first with increasing and then with decreasing increments.

Testing the Drum Motion of Indicators. An apparatus for determining the relative accuracy of the drum motion of indicators as regards uniform tension in the cord for a given speed is illustrated in **Fig. 123.**

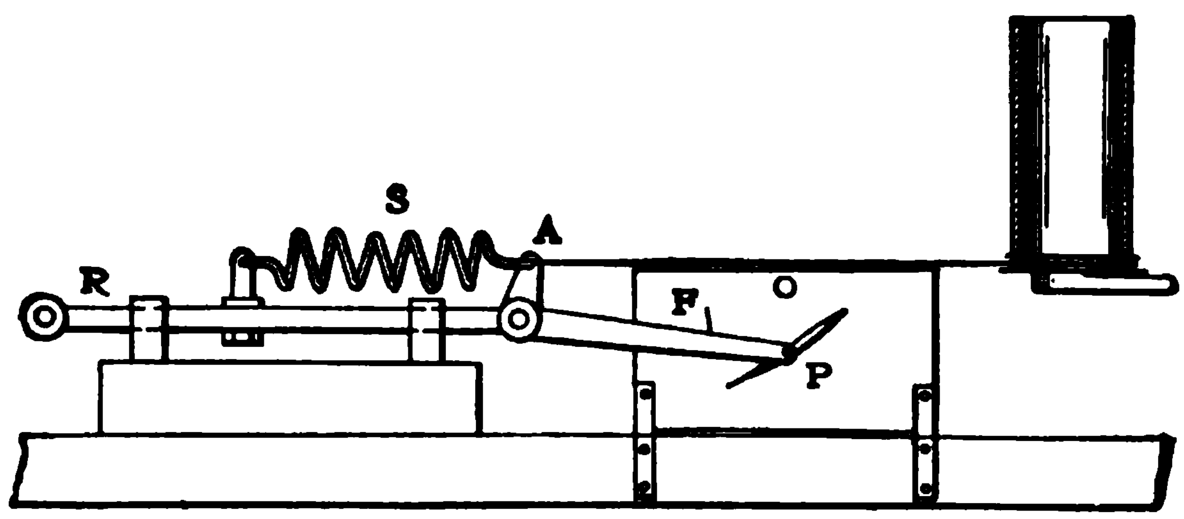


FIG. 123.—Brown's Apparatus for Testing Drum Motion.

This device, known as Brown's, consists of a rod **R,** which is made to take the same movement as the end of the cord in a reducing motion by being attached through a connecting rod to a crank pin on a disk like the face plate of a lathe. At the other end this reciprocating rod is attached to a bell-crank lever **F,** of which the outer end **P** carries a pencil for making a diagram on a card attached to a vertical frame **O.**

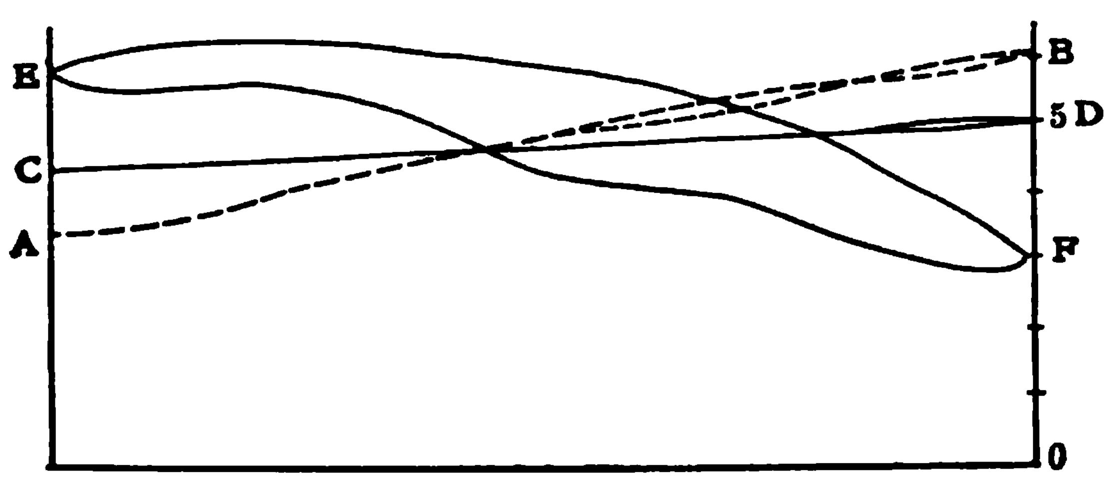


FIG. 124.—Diagrams taken from Apparatus for Testing Drum Motion.

The short end **A** of the bell-crank is connected to a helical spring **S** and the other end of this spring is attached to an arm fitted on the rod **R.** To the end of the spring at **A** the indicator cord is attached, being of the same length as when in use on an engine. Now when the reciprocating rod **R** is in motion and the tension in the spring **S** is uniform the pencil at **P** will describe a horizontal line. If, however, the tension in the indicator cord varies and consequently also the tension in the drum spring is not uniform, the pencil will describe a closed curve. Examples of such curves are shown in **Fig. 124.** Curve **AB** was obtained when the

apparatus was moving very slowly, EF when operating at about 700 revolutions per minute, and CD when the speed was about 250 revolutions per minute. The latter speed is obviously the one for which the stiffness of the spring in the indicator drum and the length and quality of the cord are most suitable.

Brown's apparatus is useful merely in showing relative results and can be depended on scarcely at all for absolute or actual values. Professor Julian C. Smallwood has devised an apparatus shown in **Fig. 125**, from which actual values can be interpreted, as shown by the diagrams in **Fig. 126**.[1] When the drum and cross-head motions are exactly proportional the diagram traced is a straight inclined line, indicating no error in drum motion. Stretch of the cord and inertia of the drum and its spring cause a lack of proportionality. As a result when these phenomena exist a closed curve will be traced. The departure of any point on this curve from the straight line joining its uppermost and lowest points indicates the error of the drum motion.

Reducing Motions or Driving Rigs for Indicators. In the case of most engines the length of the stroke is very much longer than the greatest possible movement of the drum of the indicator. It is therefore necessary to provide some means called a reducing motion, which produces shorter movement, but which at every instant corresponds exactly with that of the cross-head. If this correspondence is not secured the length of the indicator diagram cannot be accurately reduced nor calculated, and the timing of the events or so-called "points in the stroke" will not be correctly represented.

The most satisfactory reducing motion or driving rig for an indicator is some form of well-made **pantograph** (**Fig. 127**) with a driving cord of fine annealed wire or a linen cord with a stranded wire core. Although unstretchable cord is to be preferred, it is not always available and ordinarily satisfactory results are obtainable with good hemp and linen cords if properly stretched in the process of manufacture.

Reducing motion and the cord or wire connections to the indicator should be so perfect as to produce diagrams of equal length when the same indicator is attached to either end of the cylinder. Tests of the reducing motion should show also a proportionate reduction of the motion of the piston at every point of the stroke; that is, there should be a fixed ratio in every position between the actual proportion of the stroke passed through and the apparent proportion measured on the indicator diagram.[2]

[1] For more complete description see *Power*, Aug. 20 and Sept. 24, 1912.

[2] To make this sort of test properly the diagram should be taken just before the engine is stopped at a point other than a dead-center, and this will serve as the basis of comparison with the measured length of stroke taken by the piston. The actual

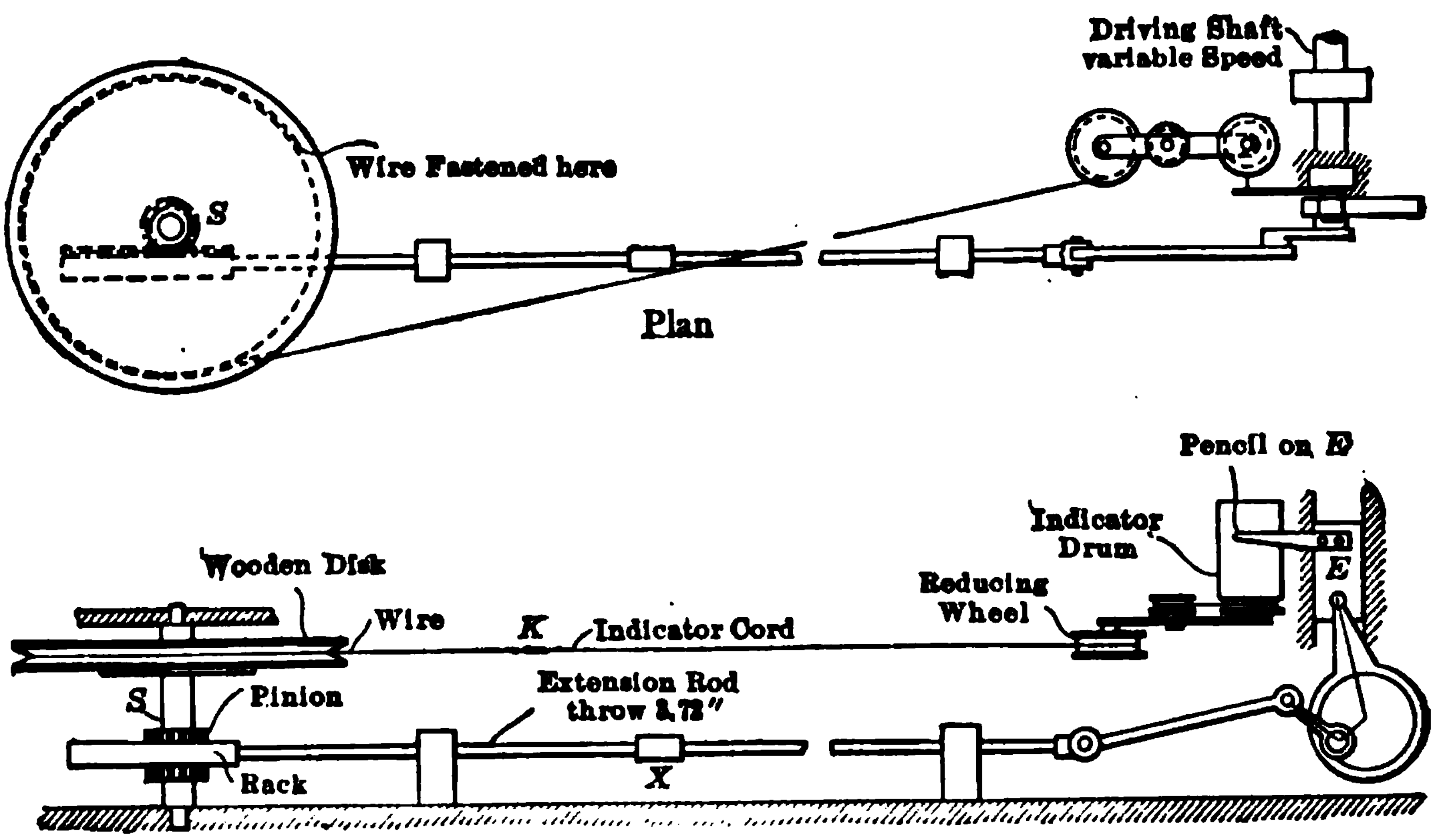


FIG. 125.—Smallwood's Drum Motion Tester.

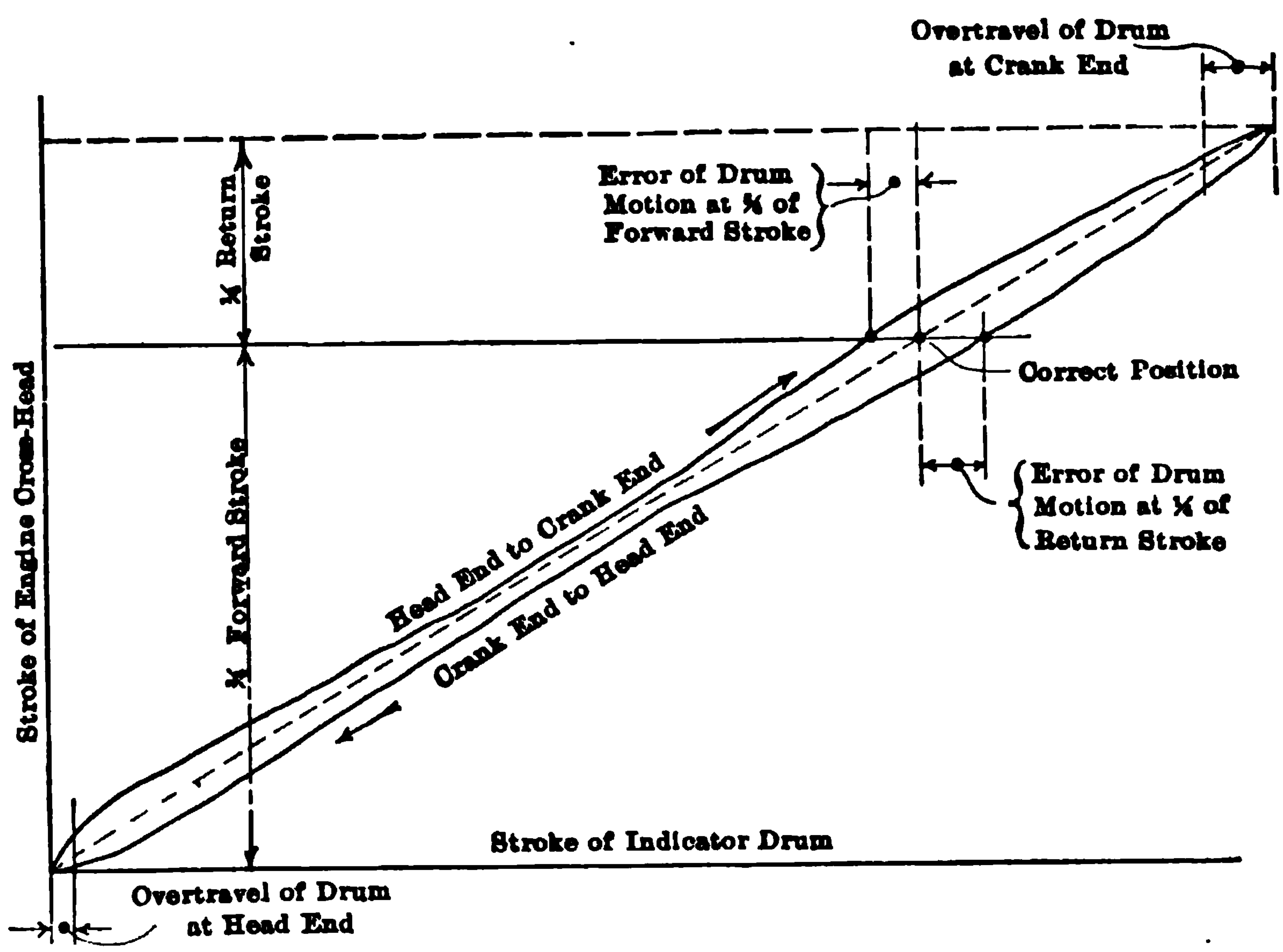


FIG. 126.—Diagram from Smallwood's Drum Motion Tester.

proportion of stroke moved through by the piston is measured by the distance the cross-head has moved compared with the full length of the stroke. Before making this measurement the slack should be taken up by putting enough steam into the cylinder to bring some pressure upon the piston, but not sufficient to start the fly-wheel.

One of the commonest and most disastrous errors made in connecting up indicators is illustrated in **Fig. 128**, where an indicator cord is shown connected directly to the cross-head **C**. Obviously near the posi-

Fig. 127. — **Pantograph or Lazy-tongs Reducing Motion.**

tion $\mathbf{C_2}$ the movement of the drum is much less for a given piston displacement than at $\mathbf{C_1}$. The difficulty is remedied in **Fig. 129** by using a pulley **P** to avoid unequal angularity of the part of the cord connected

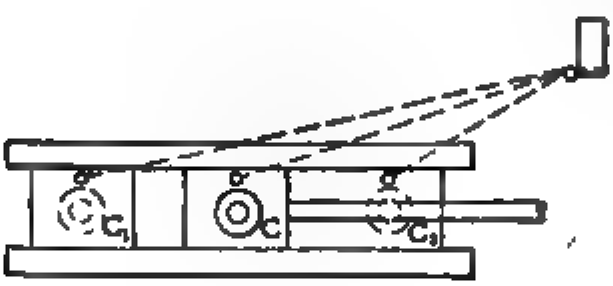


Fig. 128. — **Incorrect Method of Connecting Indicator Directly to Cross-head.**

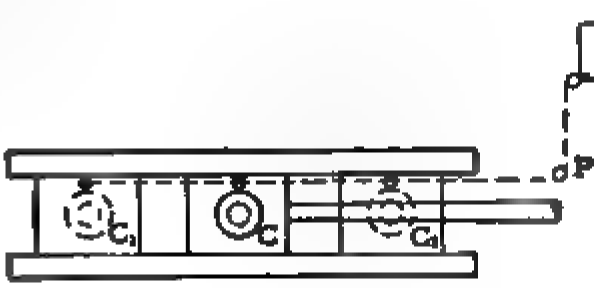


Fig. 129. — **Correct Method of Connecting Indicator to Cross-head.**

to the cross-head. Similarly unequal angularity in a marked degree of the part of the cord **connected to the reducing motion** must be avoided for the same reason. The cord must leave the reducing motion as nearly as possible in a line parallel to the axis of the cylinder.

Pipes and cocks leading from the clearance space in the cylinder to the indicators should be as short and direct as possible. Except where no other device is practicable the use of a three-way cock and a single indicator for a double-acting engine (**Fig. 161**, page 139) is not considered good practice. An indicator should be provided for each end for accurate work. The two indicators can be usually connected up to a single reducing motion in some such way as shown in **Figs. 130** and **131**.

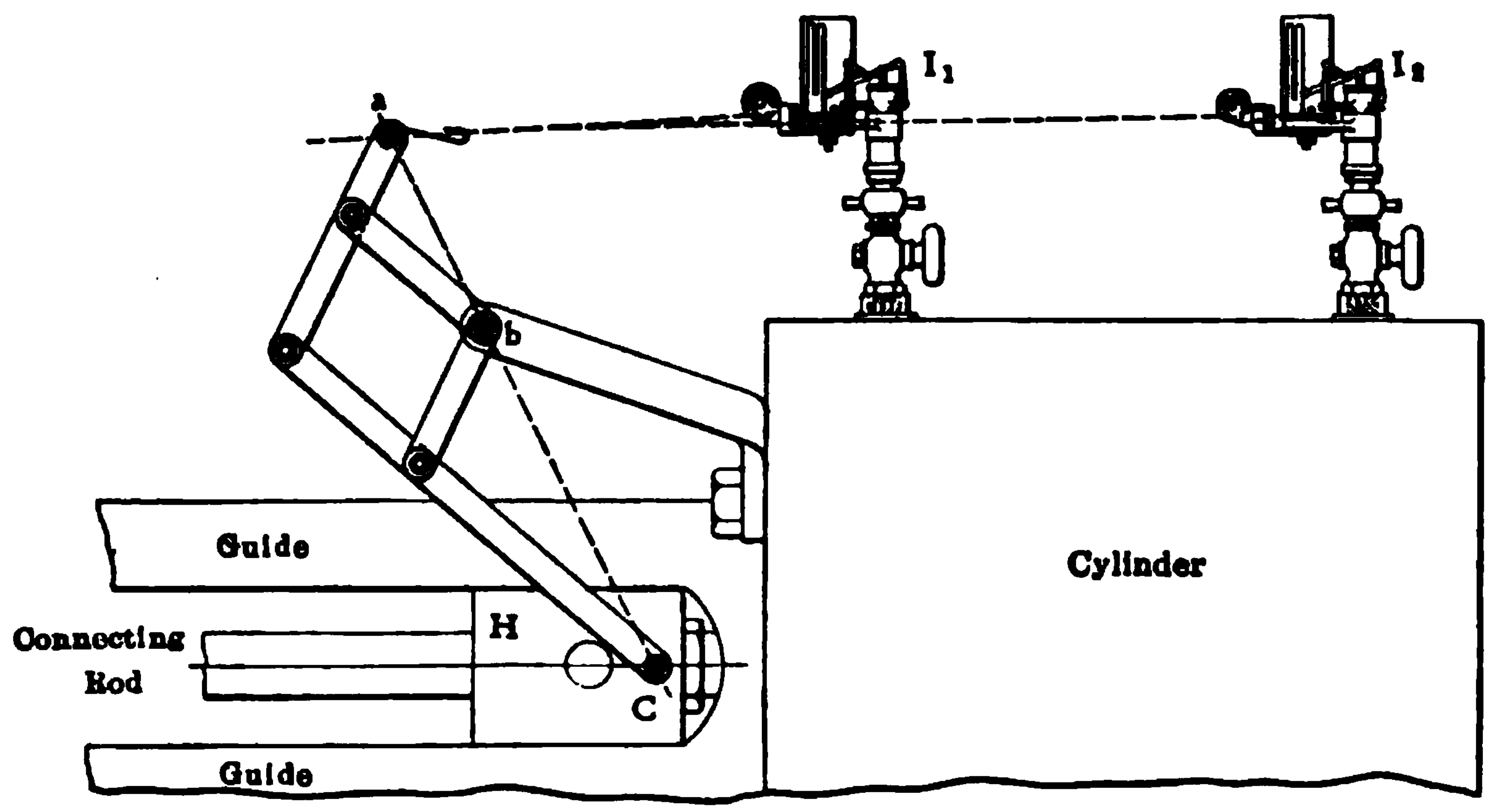


FIG. 130.—Simple Pantograph.

Error produced by the use of three-way cocks is usually in increasing the area of the indicator diagram, due to the tardiness of the indicator in responding to the changes of pressure.

One of the simplest forms of reducing motions is illustrated in **Fig. 132**. This device is pivoted at one point **A** to a pedestal supported on the frame of the engine, and has a link **BH** connected to the cross-head. The indicator cord rides in a circular arc **CD**, proportioned to give the

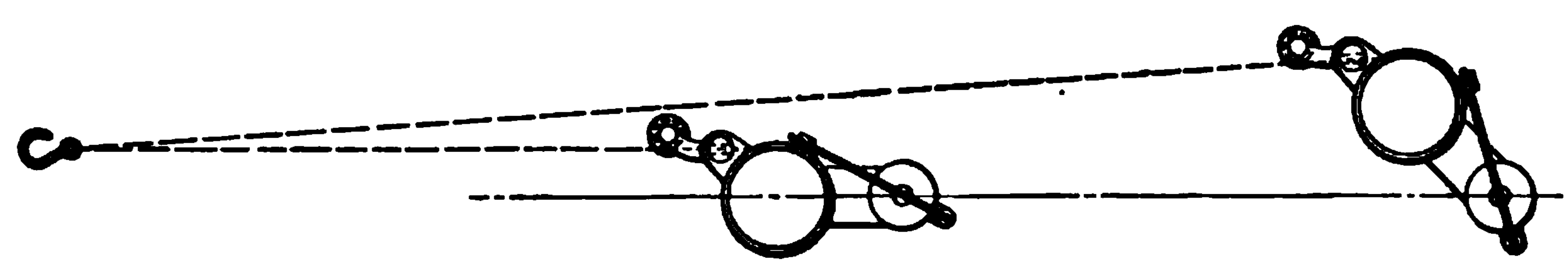


FIG. 131.—Approved Method of Connecting Two Indicators to One Reducing Motion.

required movement to the drum of the indicator. Although this arrangement does not give an exact reproduction of the movement of the cross-head, yet if the pendulum **AB** and the cross-head are simultaneously at the middle of their strokes and the position **B** is as far below the horizontal drawn through **H** (**Fig. 133**) as the extreme positions **F** and **F′** are above it, the error is insignificant. An improved type of this device is shown in **Fig. 134**, in which the cord rides in a groove on

the circumference of a quadrant pulley. By attaching the pendulum to the quadrant pulley by means of a suitably designed "slip" joint or clutch, the pendulum can be disconnected from the quadrant so that the segment and the indicator cord will be moved only when the indicator diagrams are to be taken.

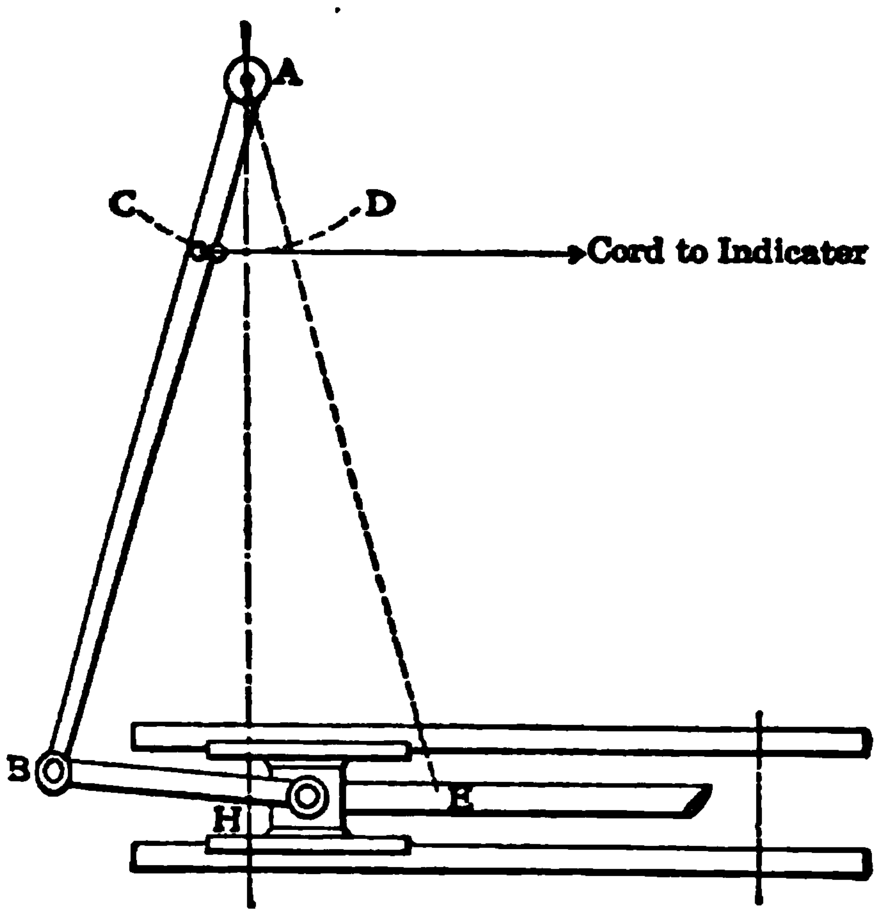


FIG. 132. — Simple Pendulum Reducing Motion.

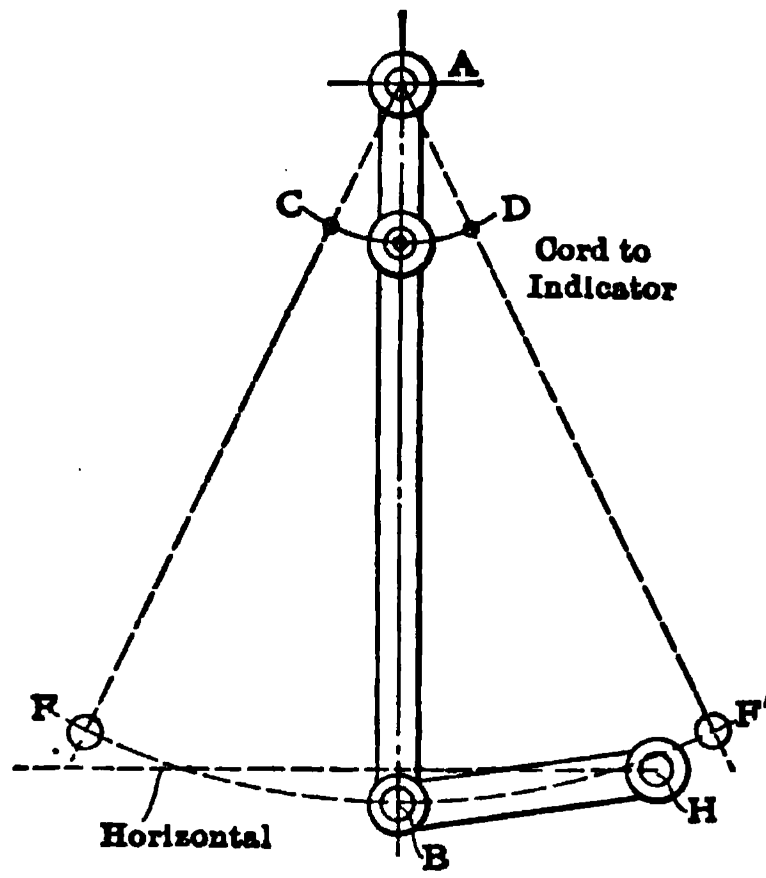


FIG. 133. — Approximately Accurate Pendulum Device.

Brumbo's Pulley is also a form of reducing motion of the pendulum type. It is illustrated in **Fig. 135**. In this device a guide pulley is placed between the indicator and the quadrant pulley. A modified and simpler device of the same kind consists of an upper portion moving in a vertical direction in a swinging tube and a lower portion pivoted directly to the cross-head. In the arrangement shown in **Fig. 135**, the pin between the segment and the indicator over which the cord passes is not absolutely essential in a temporary rig where no great accuracy is expected, as without it the change of angularity of the cord from one end of the stroke to the other is small.

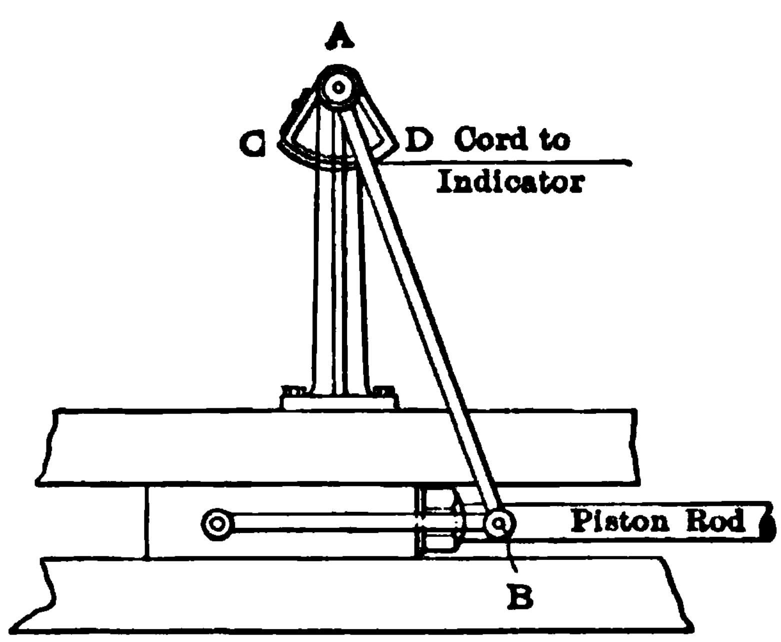


FIG. 134.—Maloney's Reducing Motion.

A very good and simple device is shown in **Fig. 136**, consisting of two rods **AB** and **BC**, together with a pin **D** for the attachment of the indicator cord. If this is made so that **AB** = **BC**, then in any position of the arms the movement of the point **D** will be exactly proportional to that of the cross-head at **C**. The point **A** is fixed to the frame of the engine.

The point **B** will always move half as far **horizontally** as **C**, because **ABC** is an isosceles triangle and the perpendicular drawn from **B** to the line **AC** bisects it. Then the **horizontal** movement of any point **D** in **AB** will be $\frac{AD}{AB}$ of that of **B** or $\frac{AD}{2\,AB}$ of the length of the stroke. Except for the very small change of the angularity of the cord as **D** goes from the

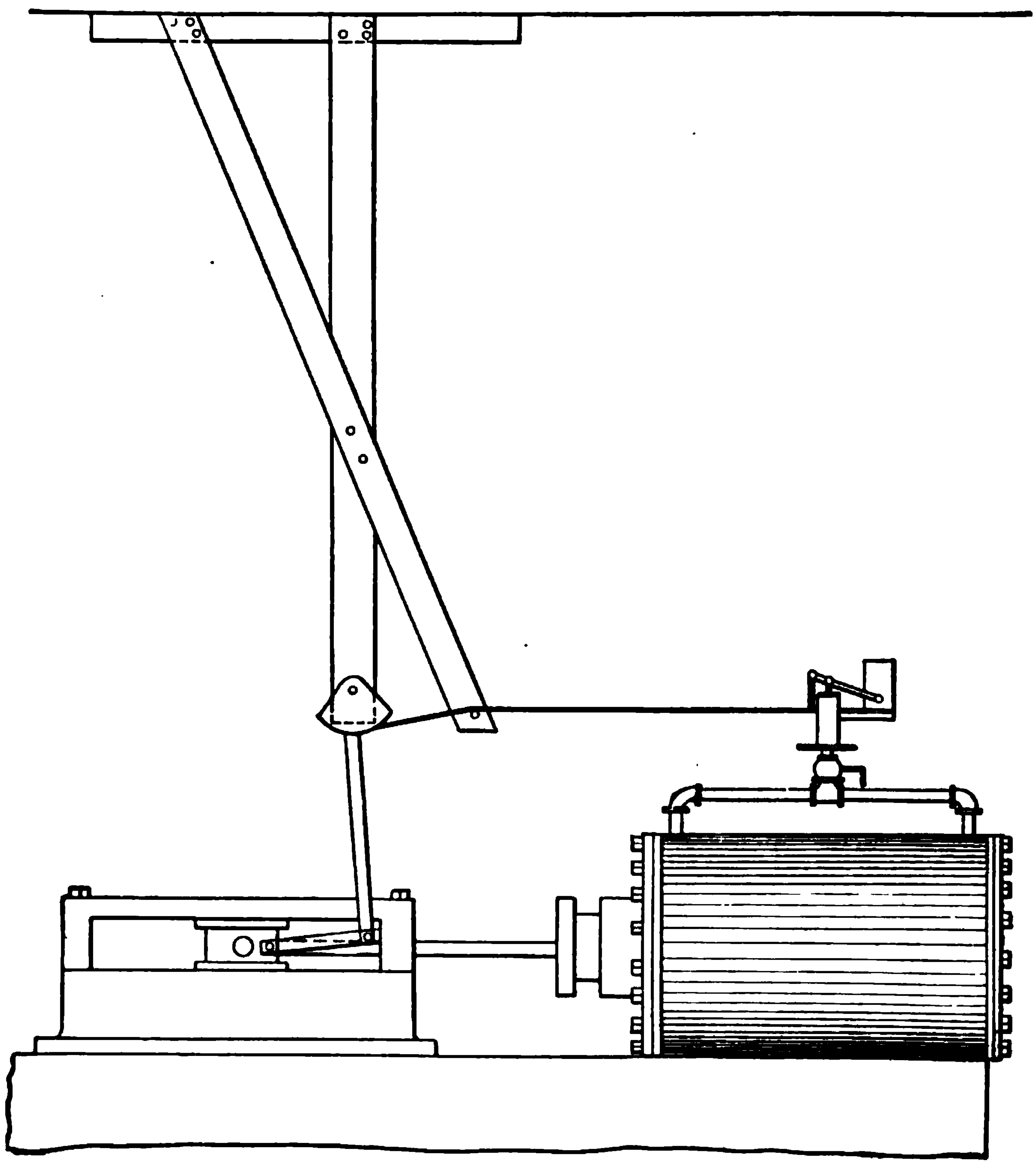


FIG. 135.—Brumbo's Pulley.

position 1 to 3, the motion of the indicator drum moved by this device will be perfect.

Figs. 137, 138, 139, and 140 are illustrations of modified forms of pantographs, sometimes known as a "**lazy-tongs.**" Because of the numerous parts of which they are composed, requiring a great number of joints, they are likely to be troublesome with high-speed engines. A plan view showing one of the methods of attachment of this device to a horizontal engine is given in **Fig.** 137. The obvious objections to the arrange-

ment as shown are that there is more than the allowable angularity of the indicator cord between the pantograph and the pulley on the indicator, and that a single indicator is connected up by a three-way cock to both ends of a double-acting engine with a long stroke.

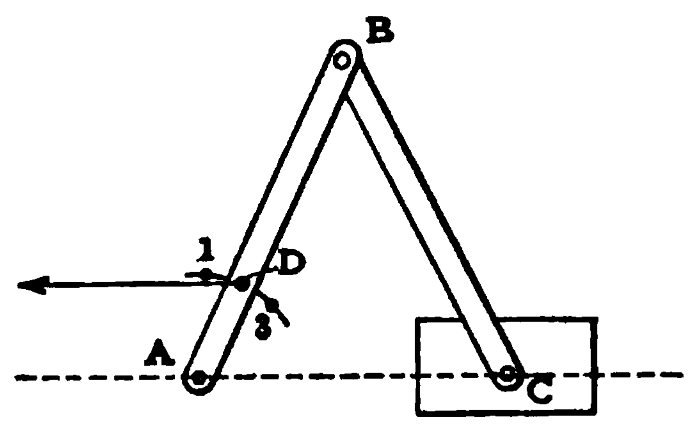


FIG. 136.—Oscillating Arm Device.

The ones illustrated in **Figs. 138, 139,** and **140** are very commonly used. They are made usually of rods of iron or of steel nicely riveted together at the joints. The indicator cord is generally attached as at **B** (**Fig. 138**) and the ends **A** and **C** of the longer rods are fastened respectively to the cross-head and to the frame of the engine. It is a necessary

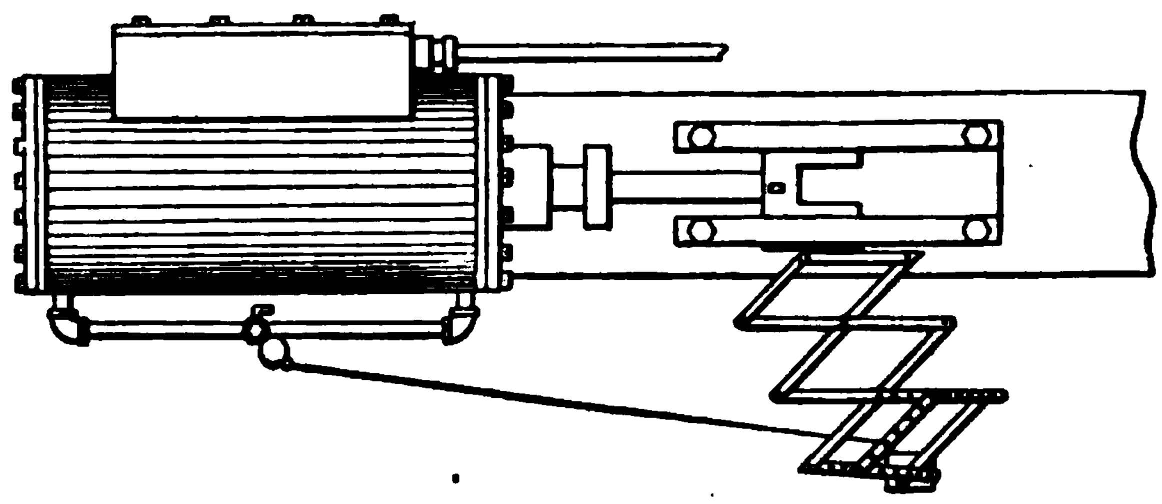


FIG. 137.—Plan View, Showing Attachment of Pantograph.

requisite that in all these pantograph types the points corresponding to **A, B,** and **C** shall lie in a straight line as shown, and **DE** must be equal

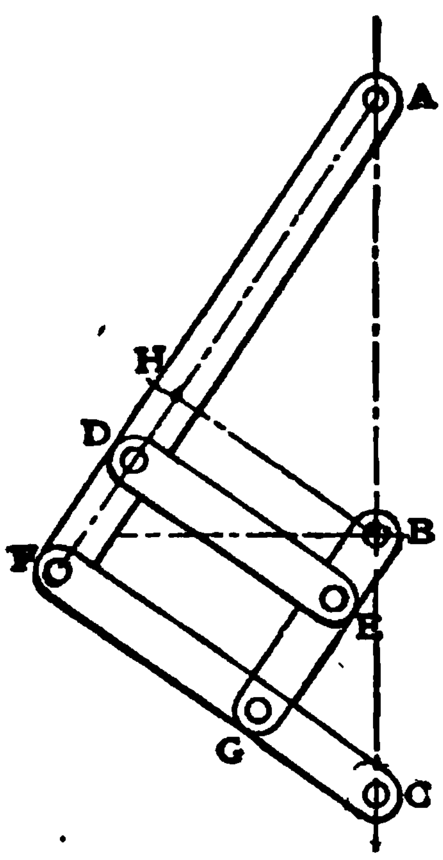


FIG. 138.—Simple Parallel Reducing Motion.

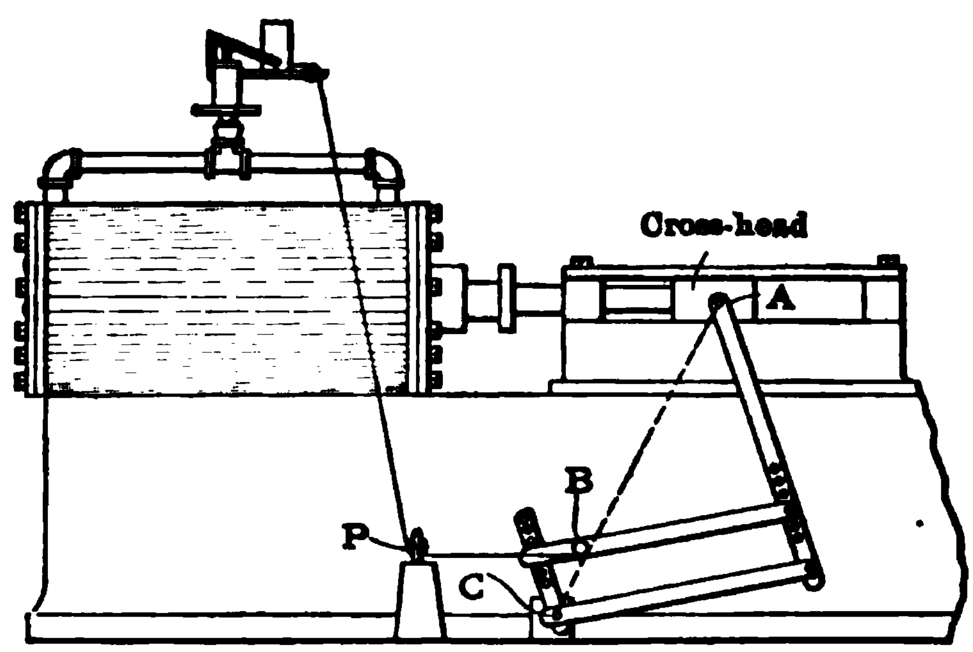


FIG. 139.—Simple Parallel Reducing Motion as Attached to a Steam Engine.

in length and parallel to **FG**. Then **AF** is in the same ratio to **HF** as the stroke of the piston is to the length of the indicator diagram.

Fig. 141 illustrates another interesting parallel motion. It consists of a rod **R**, moving in a slide **S**, parallel to the piston rod. A link **BD** is attached to the slide **R** at **B** and to **CE** at **D**, while **AE** is fastened at one end to the cross-head. In this case again if **A**, **B**, and **C** are in the same straight line, then the following relation holds: **AE : BD** and **CE : CD** as

FIG. 140.—Simple Pantograph Attached to an Air Compressor.

the stroke of the piston is to the length of indicator diagram. The cord is hooked on a pin at **H**. It is sometimes desirable to have a separate pin for each indicator used.

It is often very convenient, especially in single-acting engines, to drive the indicator directly from the main crank shaft. A device of this kind often found in practice but which does not give a true diagram is shown in **Fig. 142**. The correct method to use in this case is illustrated in **Fig. 143**, applying a small crank mechanism. For absolute accuracy the ratio of the length of this connecting rod **ab** to the length **ao** of the "equivalent crank" should be the same as the ratio of the length of the connecting rod of the engine is to the length of the engine crank.

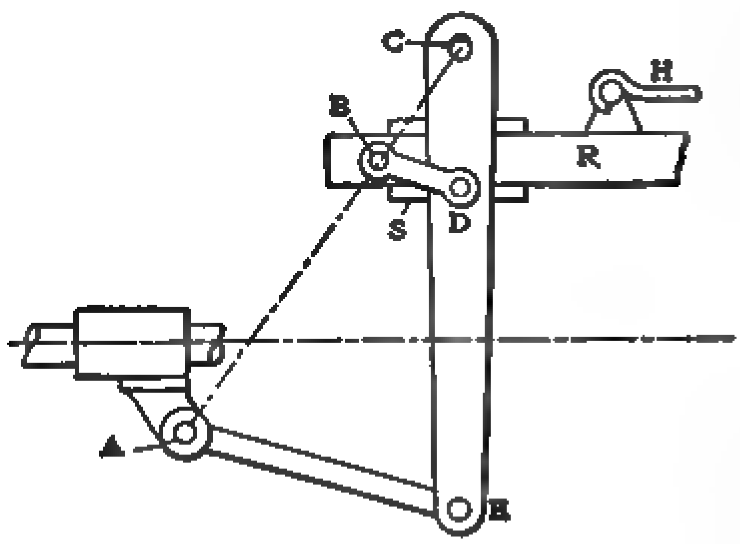


FIG. 141.—Sliding Type of Parallel Reducing Motion.

The eccentric device (**Fig. 144**) is of course the same in principle as the crank mechanism. The only advantage for the latter is that it can be conveniently adjustable for a number of engines if the eccentric **E** is made in two halves with bushings suitable for adapting it to shafts of different diameters. The line of greatest eccentricity must, of course, be set exactly parallel to the engine crank (and in the same direction).

This last statement applies also to the devices shown in **Figs. 142** and 143.[1]

Fig. 145 shows a device of the same class as the last two but suited only for use on large engines. It consists of an eccentric disk **E**, set in

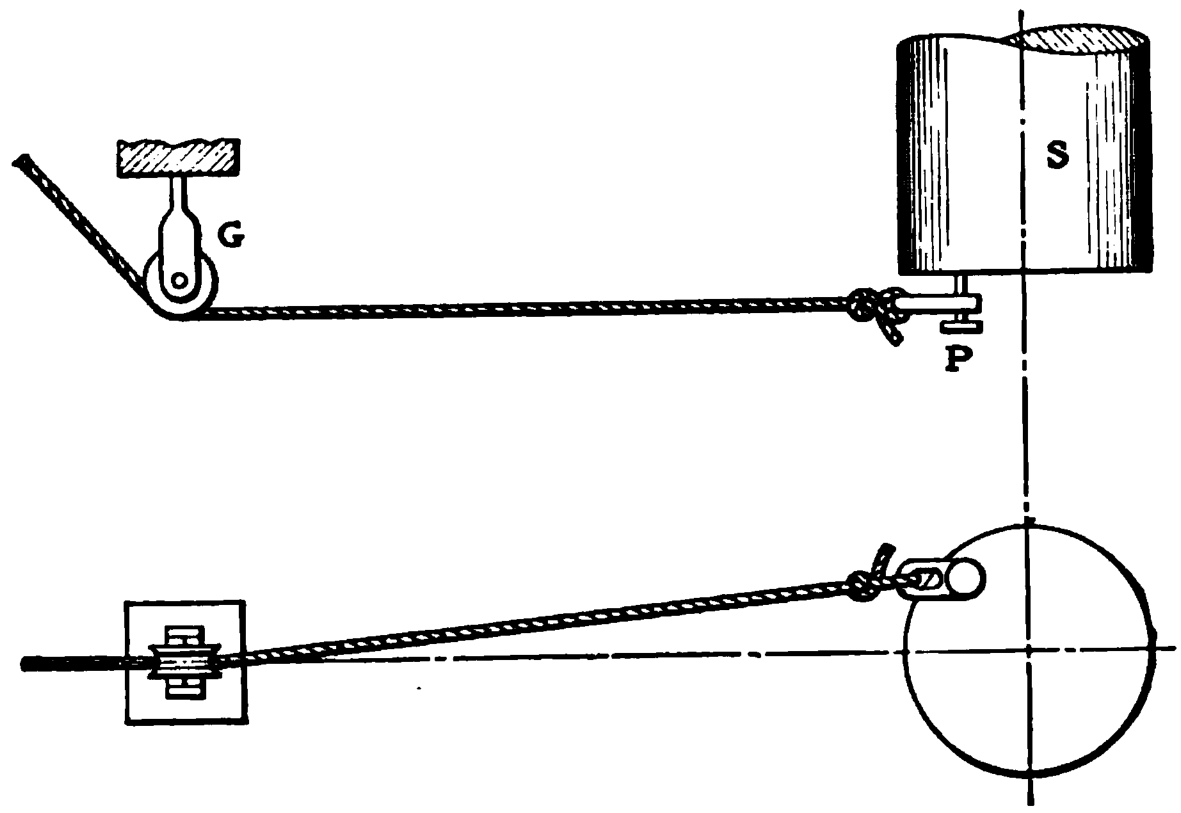


FIG. 142. — Inaccurate Crank Shaft Drive.

the manner described in the last paragraphs on the crank shaft of the engine. A side rod **T** has at one end a circular roller **R**, and at the other end a finger **F** for the attachment of the indicator cord **C**. Guides **G, G**

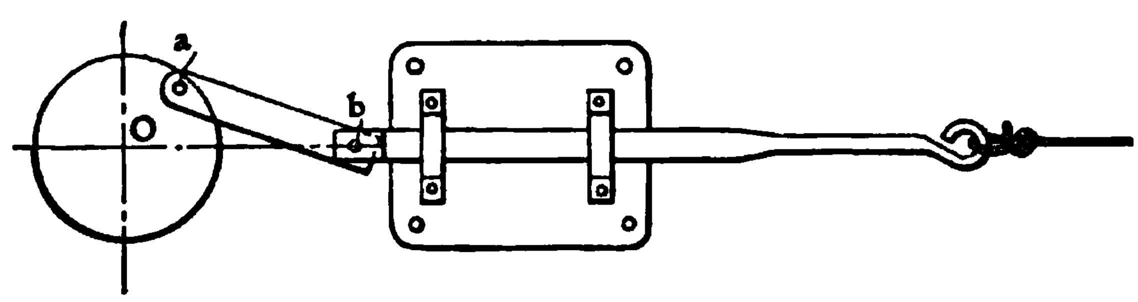


FIG. 143.

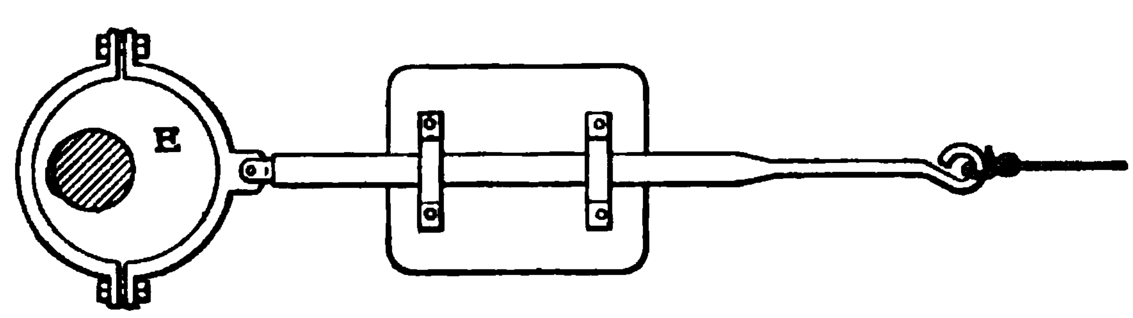


FIG. 144.

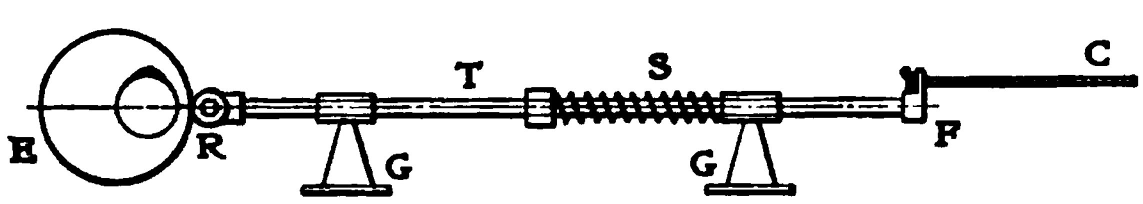


FIG. 145. — Examples of Accurate Crank Shaft Drives.

[1] The error introduced by using the plain pin device is minimized by using a wire instead of a string up to the point of attachment of the hook on the cord attached to the drum of the indicator, and making the length from the pin **P** (**Fig. 142**) to the guide-pulley **G** as long as possible.

support this rod and a spring **S** by its compression keeps the roller in contact with the eccentric. This arrangement is accurate when the following relations are satisfied: Let **L** be the stroke of the engine; **M** the length of the connecting rod; **l** the required movement of the drum of the indicator, r_1 the radius of the roller **R**, r_2 the radius of the eccentric disk **E**, then

$$\frac{L}{C} = \frac{l}{r_1 + r_2};$$

or the radius of the roller must be

$$r_1 = \frac{Cl}{L} - r_2.$$

For slow-speed engines this device is generally quite satisfactory as it permits using a short string. The spring **S** must be strong enough to

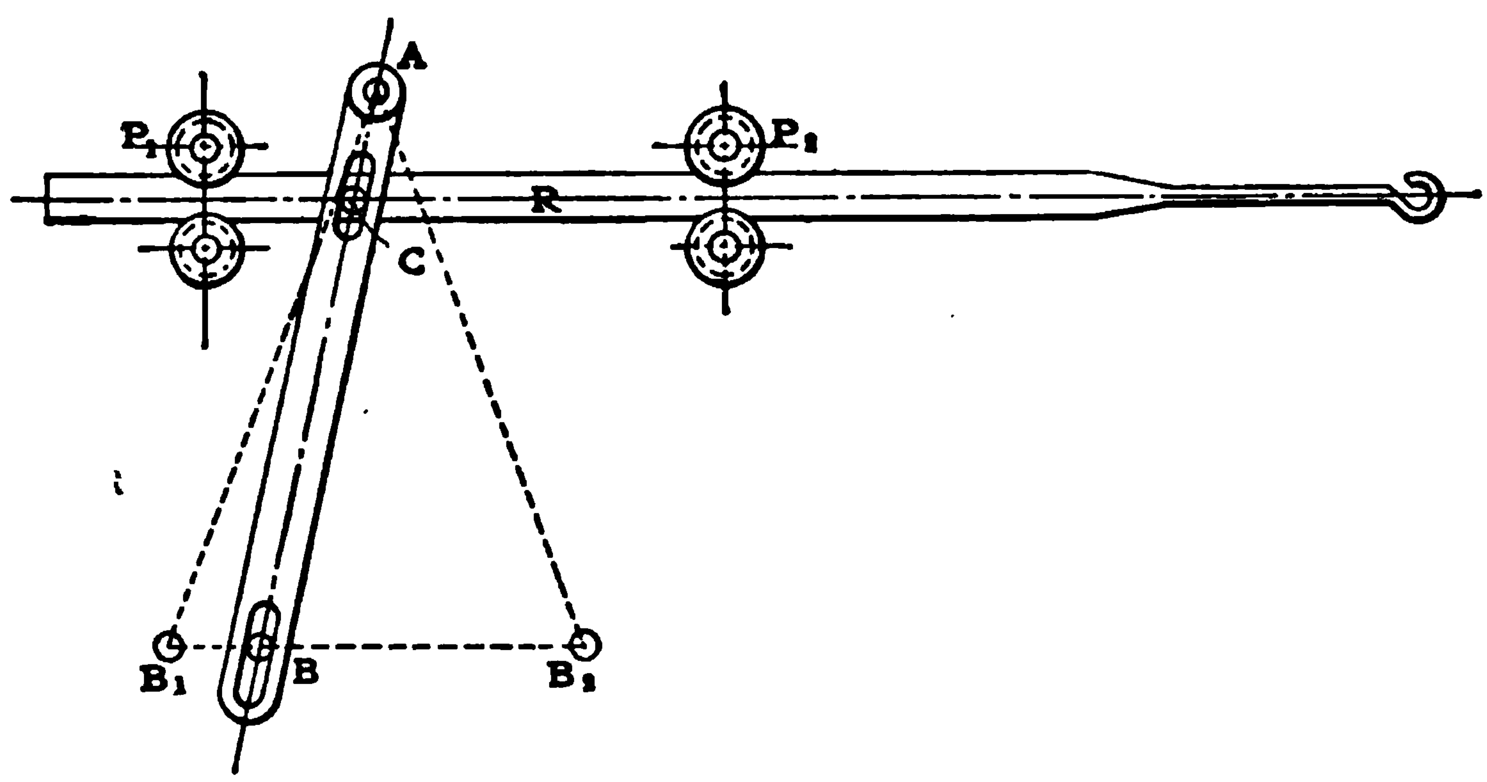


FIG. 146.—Link and Rod on Rollers.

overcome quickly both the frictional resistances and inertia effects of the reciprocating parts of the whole mechanism and the resistance of the springs in the drums of the indicators.

One of the most accurate and generally satisfactory of all the types of reducing motions is shown in Fig. 146. It consists of a pendulum-lever **AB** pinned to a pedestal or bracket attached to the engine frame in the usual way. This lever has two slots, one at the bottom for a pin **B** attached to the cross-head of the engine; the other near the top is for the stud **C** attached to a light rectangular rod **R** supported on pulleys P_1 and P_2. In this way the reciprocating movement of the piston is given directly to the indicator drive, reducing to a minimum the error due to the stretching of indicator cords. This device is especially applicable to steam pumps (either fly-wheel or duplex types), steam-driven air compressors, tandem steam engines, etc., where there are two or more cylinders in line.

One of these crank shaft devices must usually be applied in the case of single-acting engines particularly where the connecting-rod is attached directly to the piston. Another method often used on single-acting engines which do not have a closed crank case is a modification of that shown in **Fig. 132.** The stud **A** is supported in the usual way by a bracket bolted to the frame of the engine. A small fixture **F** (**Fig. 147**) provided

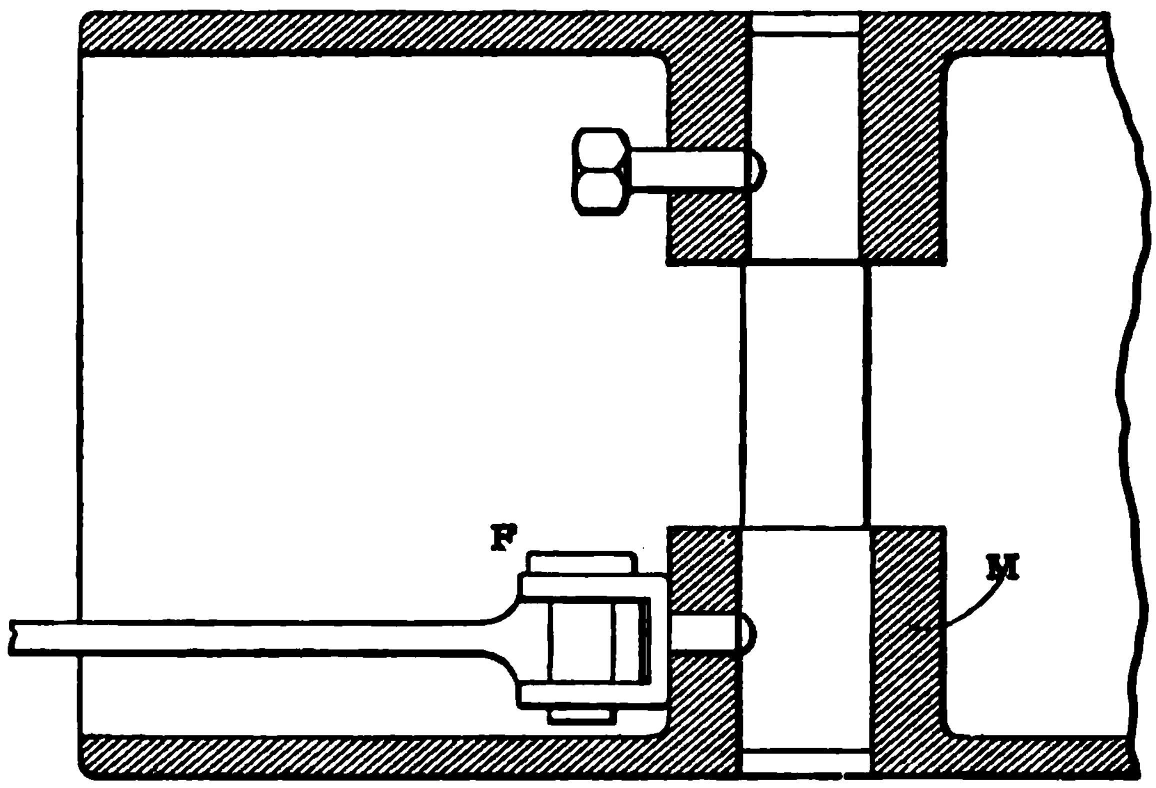


FIG. 147. — Device for Single-acting Engines.

with holes in the sides for the insertion of a small pin is attached, usually screwed, to the hub **M** of the piston. The swinging arm **BH** (**Fig. 132**) when pinned to the fixture **F** transmits freely the motion of the piston to the arm **AB**. Since both the point of attachment of the indicator cord and the point **B** move in arcs of circles about the same center **A**, the indicator drum will be moved almost in exact coincidence with the

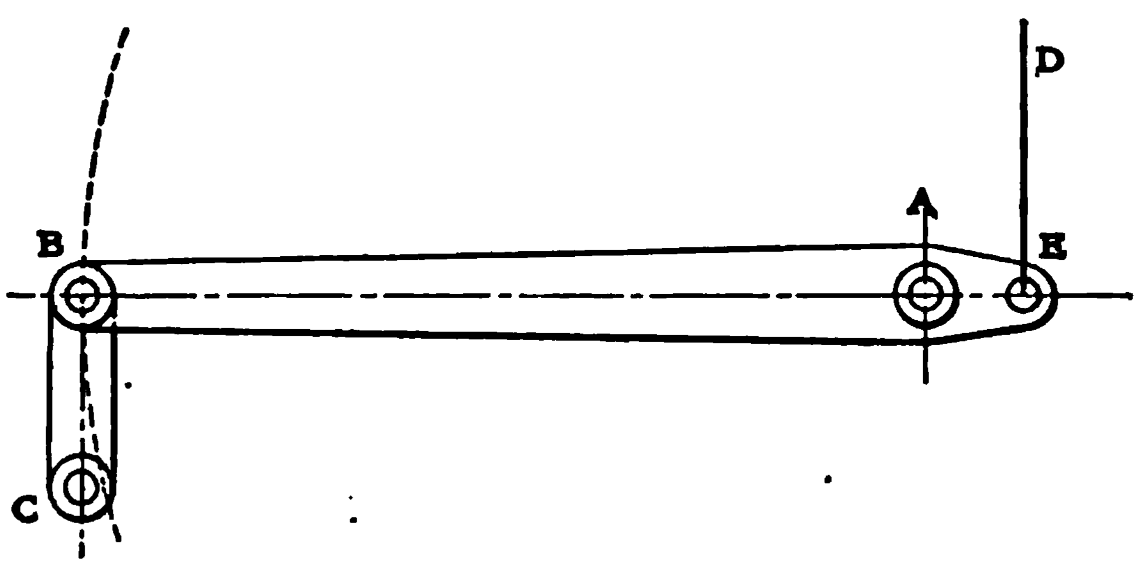


FIG. 148. — Device for Vertical Engines.

motion of the piston. The portion of the indicator cord attached to the arm **AB** should be moved as nearly as possible parallel to the line of the stroke of the engine for the best accuracy.

An indicator reducing motion suitable for use on large vertical engines is shown in **Fig. 148.** In construction it is very simple, consisting of only two levers. The longer one **BE** is supported on a pin in a bracket at-

tached to the frame of the engine. The other lever **BC** is pinned to the cross-head at **C**. The indicator cord is fastened to the mechanism at **E** and should be arranged to be parallel to the line of the stroke of the engine. When a reducing gear of this kind is used the indicator cords would be very long in many cases, and a steel wire should be substituted for the ordinary indicator string. One end of the wire should be attached in the usual way at **E**. At the other end where it is to be hooked to the strings on the indicators, a small and light spiral spring should be attached by means of a short string. This spring when connected to some stationary part of the engine near the cylinder serves to keep the wire taut when the indicator cords are unhooked. At times when a spring is not available a long rubber band can be substituted.

Reducing Wheels which consist simply of a large and a small pulley attached to the same axis, are coming into more or less general use. A typical arrangement is illustrated in **Fig. 149.** Pulleys **D** and **D′** are

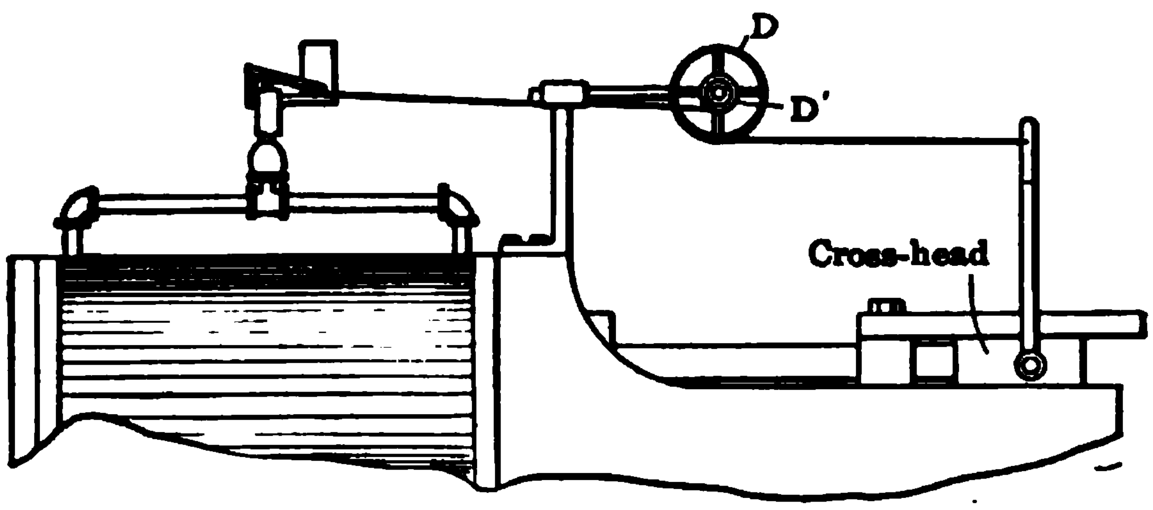


FIG. 149.—Reducing Motion of Concentric Pulleys.

usually connected by a sliding sleeve so that they can be disconnected when indicator diagrams are not being taken.

Fig. 150 is a device **for engines with long strokes.** **A** and **B** are fixed ends of cord wrapped around a pulley **D**. The indicator cord **g** is attached to a small pulley **D′** and passes around a guide pulley **G**. **D** and **D′** are attached to the cross-head **C**. Then diameter **D** ÷ diameter **D′** = stroke of piston ÷ by the difference between stroke of piston and length of card.

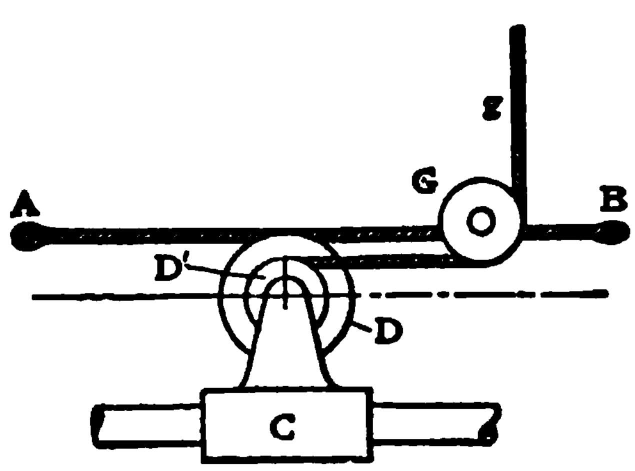


FIG. 150.—Armand Stewart's Reducing Motion.

Reducing wheels are not infrequently made for attachment directly to the indicator, as illustrated in **Figs. 151, 152** and **153.** The first shows the **Crosby** reducing-wheel attachment, the second a similar device for the **Tabor** indicator and the third an English device with three pulleys. These are designed for direct attachment to the drum of the indicator. In **Figs. 151** and **152** the cord on the wheel comes from the cross-

FIG. 151. — Crosby Indicator and Reducing Motion Attachment.

FIG. 152. — Reducing Motion Attachment for Tabor Indicator.

head or from a bracket on it, and gearing drives the drum of the indicator. The ratio of the diameters of the wheels gives the reduction ratio. Each reducing wheel is provided with a "nest" of annular rings to be put on the small wheel to increase its diameter, so that various reductions are easily arranged to suit the stroke of the engine. The device shown in Fig. 151 is started and stopped by turning the handle 14 and moving the clutch 25. The one in Fig. 152 is started and stopped by turning the knurled nut along side of the wheel carrying the cord. A spring is located at the center of this wheel for bringing it back on the inward stroke.

Fig. 153. — Three-pulley Reducing Device.

An arrangement of two indicators I_1 and I_2 to be driven from a single reducing wheel **R** is shown in Fig. 154. The connection to the crosshead **C** is by a bracket **B**. It is best in most cases to have a reducing wheel for each indicator in use but this arrangement will often be very serviceable when, for example, one of the reducing wheels has broken down.

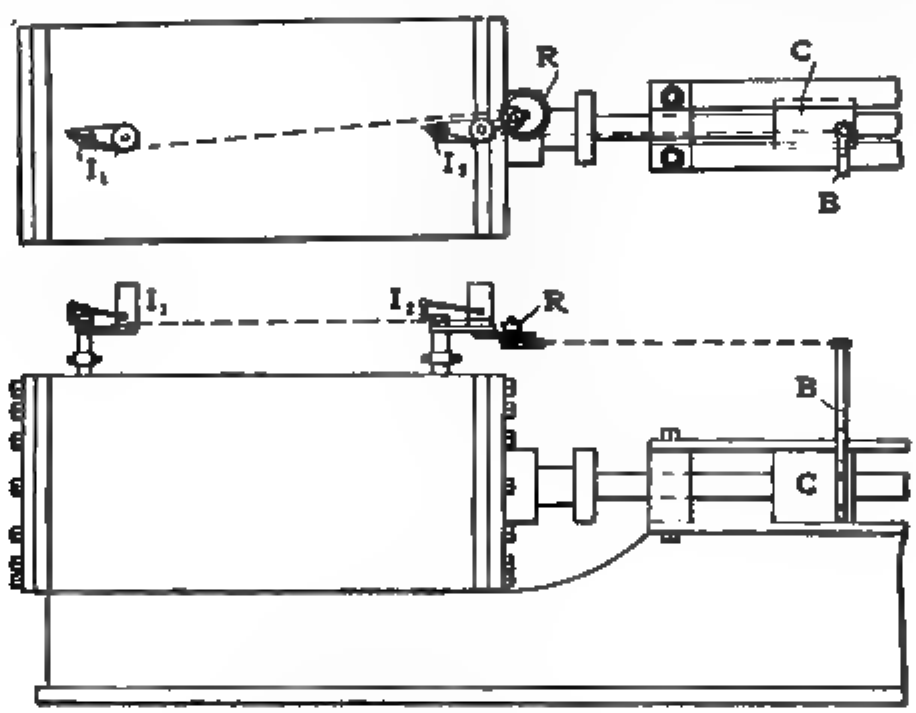


Fig. 154. — Single Reducing Wheel Driving Two Indicators.

Some indicators are provided with "**detents**" which are devices for starting and stopping the drum. For stopping the drum they operate by engaging a pawl into teeth on the circumference of the drum near the bottom (see Fig. 107). Obviously the pawl must engage when the cord is pulled out to near the end of its stroke. After engaging the pawl the cord will be like y to flap about on the return stroke, catch on something, and be broken on the next outward stroke. The best way to prevent

this is to connect the string to a helical spring or rubber band to keep it taut.

In engine testing for long periods it is desirable to save wear of reducing wheels as much as possible by disconnecting the cord connecting the pulley with the cross-head during the intervals between the taking of diagrams. Hooks like those shown in **Figs. 155** and **156** are very convenient for this purpose, particularly if attachment can be made to pins or rods fastened to the cross-head. The hook shown in **Fig. 155** is intended to be held between the thumb and finger at about an inch from either end of the stroke, so that the pin or rod on the cross-head strikes the straight part of the hook, and immediately the pin or rod will be caught in the hook as shown by the dotted lines. The one shown in **Fig. 156** is also frequently used. Both hooks are intended to be pulled off the pin or rod when the cord is to be disconnected. Unless some special form of hook similar to the ones described is used it is difficult to connect up the cord when a diagram is to be taken. Provision should also be made for preventing a disconnected indicator cord from being broken by getting tangled in the moving parts of the engine.

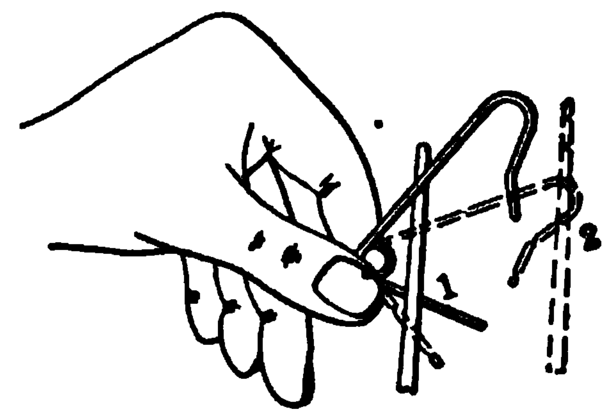


FIG. 155. — Trill's Hook for Indicator Cord.

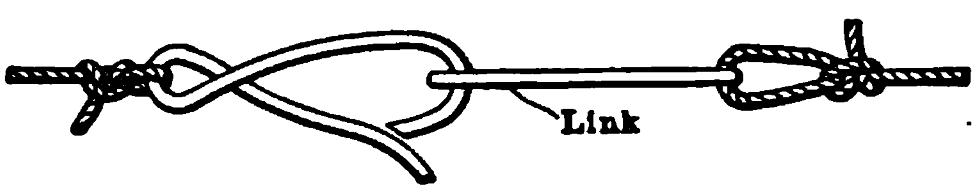


FIG. 156. — Simple Hook for Indicator Cord.

Such difficulties can usually be avoided, and easily with such devices as **Fig. 155** by continuing the cord from its point of attachment to the reducing motion closely past the indicator drum to a pipe or simple bracket that may serve as a stationary support. Between the indicator and this support a spiral or helical spring of light wire or a heavy rubber band is inserted in the cord. By this means the cord will always be kept taut and in motion when disconnected from the reducing wheel or from the indicator, as the case may be. If a ring is attached to the cord close to the reducing wheel and between it and the cross-head, it will be continually in motion but it will be very easy to hitch into it the hook on the cord attached to the reducing motion. This method is particularly recommended in every case where wire is used instead of cord.

Relative Accuracy of Indicator Motions. Pantograph types like **Figs. 127, 130** and **137** to **139** give usually the best accuracy on small

engines, if constructed accurately and are used with short strings. The most important of the disadvantages is that invariably after a time they get shaky in one or more of the various joints, causing lost motion in the drive and many breakdowns in the indicator rigging. In reducing motions like **Fig. 146** such difficulties are almost entirely eliminated and at the same time, if the pins fit the slots nicely the movement is accurate. It is, however, expensive and somewhat cumbrous. Pendulum types, like **Figs. 132** to **134**, are accurate enough for most work. The length of the pendulum should in either type be not less than one and a half times the length of the stroke. Brumbo's pulley (**Fig. 135**) is shown as an example of what can be improvised with the assistance of a carpenter and willingness to get things done. The "pin and cord device" shown in **Fig. 142** should be used for only very rough work.

Types using concentric pulleys, like **Figs. 149, 150, 151, 152,** and **153,** although suitable and quite serviceable for use on low-speed engines, are not accurate enough for good work on even moderately high-speed engines, and are always likely to make trouble on account of the indicator cords and wires getting twisted and torn. The inertia of the pulleys at high speeds tends to cause them to overrun at the ends of the stroke. Of the last mentioned, those having no means for mechanically starting and stopping the motion of the drum are also objectionable because of the difficulty usually experienced in hooking and unhooking the cord.

The crank shaft drives illustrated by **Figs. 143, 144** and **145** are to be used only when one of the better methods is impracticable, as for example in the case of single-acting engines. The pendulum type shown by **Fig. 147** can be used also on some single-acting engines and is to be preferred to the others mentioned. It cannot be used on types of engines having a closed crank case like some small vertical steam engines, most automobile engines, and two-cycle gas engines.

All reducing motions should be designed to operate with a minimum number of guide pulleys.

Errors in Indicator Diagrams and in Calculations of Indicated Horse Power. Taking indicator diagrams from steam or gas engines is regarded by most engineers as a very simple and commonplace operation, and consequently it is not unusual to find inexperienced persons placing great reliance upon indicator diagrams and the calculations therefrom, when as a matter of fact the conditions under which the diagrams were obtained make them in error possibly from five to ten per cent. Except in the hands of an expert the engine indicator, even of the best makes, cannot be relied on to give the mean effective pressure in the engine cylinder with a smaller error than two or three per cent. Prin-

cipal sources of errors found in indicator practice will be discussed in the following paragraphs:

1. **Errors in the Reducing Motions.** The theory of the application of the engine indicator requires that the drum shall move in synchronism with the piston of the engine, following accurately its changing speed from one end of the stroke to the other. Errors in the application of this principle are usually found in the construction of the reducing motion. A familiar example of such errors is shown in **Fig. 142** (see page 129) showing the drive for the drum of the indicator taken directly from a pin on the crank shaft of the engine. Another common error is in the application of parallel motion drives like **Figs. 138** to **140,** when the points marked **A, B** and **C** are not in line; in other words, so that a straight line will not pass through the three points. Stretch in the string, particularly if it is long, is a fruitful source of error. If the string stretches only a little so that it does not bring the drum up against one of the stops, this error is difficult to observe; but when the cord becomes so long that the diagrams drawn are not of the normal length, the effect can be detected by the knock of the drum against the stop and also by the distorted diagrams, of which **Fig. 157** is an example. The full lines show the kind of diagrams that are obtained when the indicator drum strikes one of the stops. The dotted lines show the parts of the diagrams that are missed.

2. **Vibrations due to Inertia.** Indicator diagrams often show irregular lines due to vibrations of the mechanism of the indicator, being particularly pronounced just after a sudden change of pressure like that occurring in a gas engine following the point of ignition. **Fig. 158** is an indicator diagram taken from a gas engine which shows such oscillations. The same sort of effect is observed in high-speed steam engines after the sudden changes of pressure that occur at the points of admission and of cut-off. The great fluctuation of pressure at admission causes the oscillation at **A** in **Fig. 159,** while the irregularities in the expansion line at **B** follow the closing of the steam ports at cut-off. Vibration effects can usually be best eliminated by using a stiffer or " stronger " spring.

It is well known that **frictional resistance will dampen out vibrations.** For this reason some engineers advise pressing the indicator pencil rather heavily on the drum to eliminate vibrations. This method, however, is not recommended, as errors due to excessive friction may be introduced which are far greater than those that can possibly be due to the oscillations shown in the diagrams.

Attributable to vibrations will be practically only those errors resulting from the incorrect measurement of the areas of the diagrams. Generally a mean line drawn through the oscillations so that the areas of the loops on one side of the mean will be equal to the areas on the other

gives the average pressure. A mean line, shown dotted in **Fig. 158,** illustrates the method. Since it is usually difficult to follow accurately with a planimeter lines with long oscillations it is usually desirable to draw the mean line in all such diagrams and follow this with the tracing-point of the planimeter.

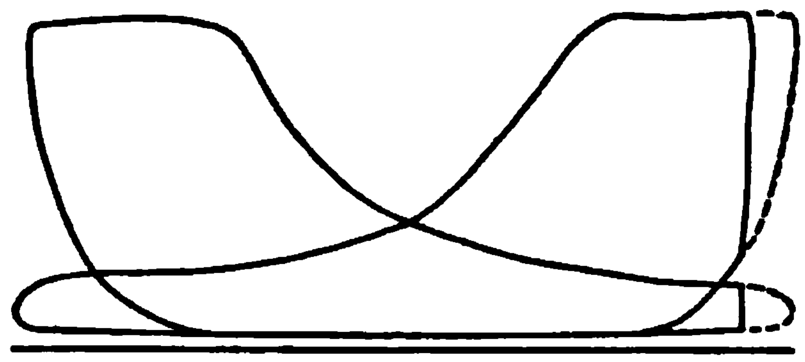


FIG. 157.—Result of Having Indicator Cord too Long.

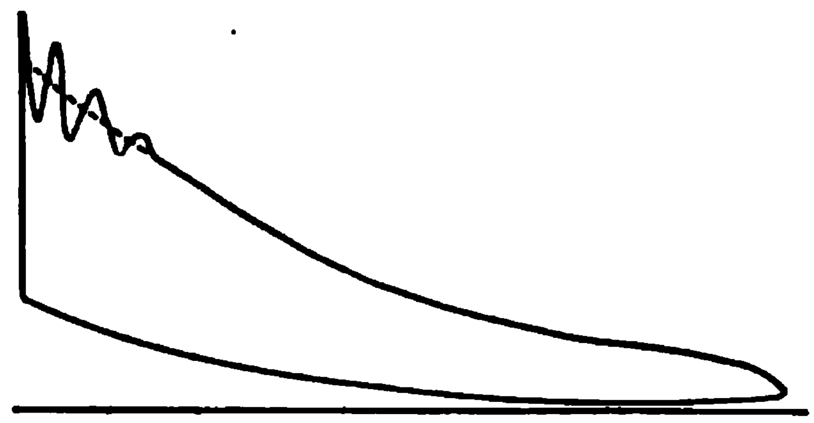


FIG. 158.—Oscillations in Diagram from a Gas Engine.

3. **Frictional Resistances.** That serious errors in indicator diagrams are often caused by excessive friction has already been pointed out particularly as regards the friction of the indicator pencil on the paper. An example of excessive frictional resistance in the indicator cylinder is shown in **Fig. 160.** Friction in this case was probably due to a "sticky" piston; that is, one that did not move freely in the cylinder of the indicator. The peculiar "step-like" appearance of the expansion and compression lines are characteristic of this fault. Irregularities in the lines

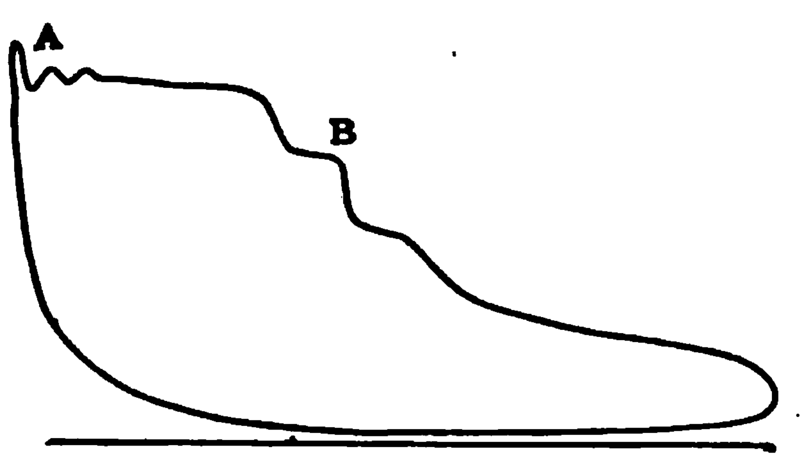


FIG. 159.—Oscillations in Diagram from a Steam Engine.

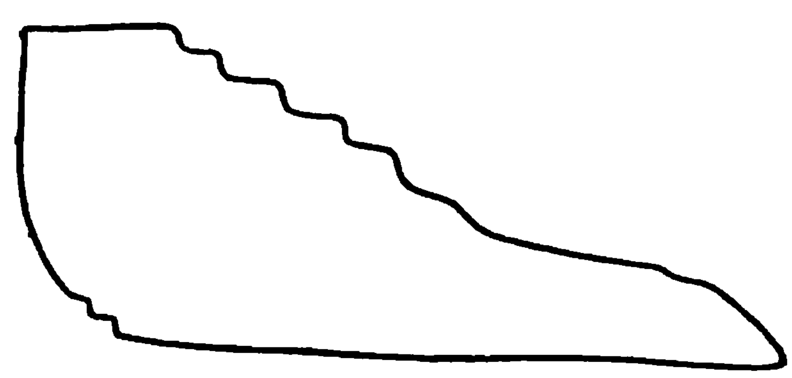


FIG. 160.—Diagram Illustrating Effect of a Sticky Piston.

of diagrams due to friction cannot be corrected as accurately as can be done for pure oscillations; because in this case the pencil does not necessarily oscillate on the two sides of the true mean line equally, but is almost continually **above** the true mean line of pressure in an expansion line, and correspondingly below when drawing the mean line of compression. In the case of distortion due entirely to pencil friction the areas balance up fairly well, but tests show that the points in the cycle are late. It is therefore very important that the pressure of the pencil-point on the paper should be carefully adjusted by the stop-screw so that it makes a **fine, light but clearly legible line.** With some types of

indicators specially treated[1] cards are provided on which a metallic pencil-point is to be used. Diagrams taken on such cards show always very fine and clear lines, but it is doubtful whether such diagrams are as accurate as those taken with a good fine black lead pencil on a card of ordinary paper. The reason for this criticism of the "prepared paper" method is that, excepting the case of the skilled operator, there is likely to be more friction in taking the diagram.

4. **Errors due to Faulty Indicator Connections to the Cylinder.** In many small power plants there is only one good engine indicator. On this account engines fitted for only commercial indicating are usually provided with a **three-way cock** (**Fig. 161**) which is used to connect either end of the cylinder with the indicator. This arrangement of piping is illustrated in **Fig. 135**, page 126. For accurate testing, how-

Fig. 161. — Typical Three-way Cock.

ever, this sort of arrangement is not desirable, as the length of the pipe connections has an influence on the accuracy of the indicator diagrams. Long pipe connections have the effect of causing the indicator pressure to lag behind that in the cylinder,[2] and may thus give a mean effective pressure two or three per cent too large, depending, of course, upon the diameter and length of the pipe connections. Pipes which are too small in diameter have very much the same effect as those that are too long. **Fig. 162** shows a distorted diagram caused by indicator connections that were much too long. The dotted line indicates the distorted diagram. A better arrangement is to screw an indicator cock directly into the end of each cylinder, and use two indicators as shown in **Fig. 130**, page 124.

The arrangement of a three-way cock and double-pipe connections should never be permitted on either ammonia or air compressors because the clearance volume in compressors is usually reduced to the lowest

[1] An ordinarily good quality of paper can be prepared for use with brass points by applying to one side of the paper a thin coat of a mixture of about one part (by weight) of zinc oxide (ZnO), four parts of water, and one-tenth part of gum arabic.

[2] W. F. M. Goss, *Transactions A.S.M.E.*, vol. 18 (1896).

limit mechanically permissible, and such pipe connections would increase the percentage of clearance enormously. For use in testing compressors and very small engines the pistons of the indicators used should not leak much because leakage affects the degree of compression and leakages of ammonia fumes are most objectionable to those operating the plant.

5. **Lost Motion in the Piston Connections and in the Joints of the Pencil Mechanism.** This is another fruitful source of errors. Fortunately, however, every engineer is usually able to get rid of lost motion due to looseness of joints without assistance.

6. **Stretching of the Cord.** The effect on the diagram of the stretching of the cord when an engine was operating at 350 revolutions per minute is shown in **Fig. 163**. The full-line diagram was taken with a wire (no stretch) and the dotted line with a good quality of prepared hemp indicator cord. Stretching of the cord is due mostly to friction and inertia of the drum[1] of the indicator. It has been shown that the mean effective pressure is proportional to the stretching of the cord caused by drum friction and that the least amount of stretch in a good indicator cord is $\frac{4}{10}$ per cent of its length. In the case of a cord three feet long, the stretching under the best conditions would be about $\frac{14}{100}$ inch, making an error of about 5 per cent in the mean effective pressure in a diagram three inches long.

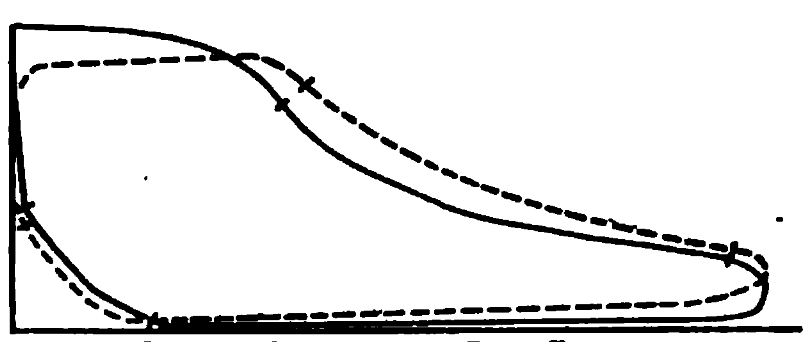


Fig. 162. — Diagram Showing Effect of Long Connecting Pipes to Indicators.

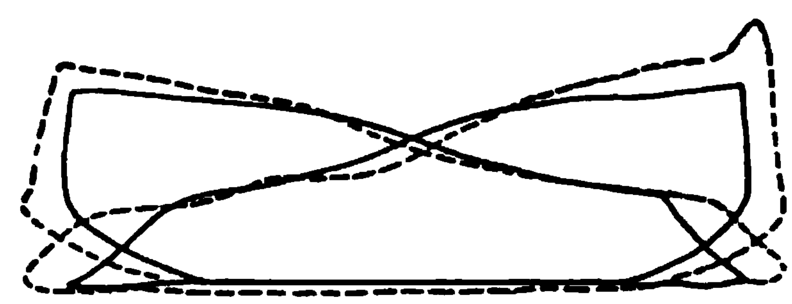


Fig. 163. — Diagram Showing Effect of Stretching of Indicator Cord.

Fortunately these errors due to inertia and friction have just opposite effects, the one making the diagram longer and the other making it shorter. The net result is apparently, according to reliable tests, that the indicated horse power even at speeds of from 300 to 400 r.p.m. is probably never in error more than 2 or 3 per cent if the indicator spring, drum spring and quality of cord are properly selected for the pressure and speed, assuming also that the piston and drum move freely and are well lubricated.[2]

[1] Professor Osborne Reynolds in *Proceedings Institution of Civil Engineers* (London), vol. 83.

[2] *Proceedings Inst. Mech. Engs.*, 1909, pages 785–798.

Diagrams obtained with the ordinary piston and pencil indicator are probably as accurate as other data in engine testing if the speed does not exceed 300 or 350 r.p.m., but at higher speeds such diagrams are regarded as of little value.[1] Automobile and aeroplane engines usually operate at 2000 r.p.m. and over at maximum power and special indicators are required. The best known is the manograph or diaphragm type. An instrument of this kind shows practically no inertia effects and there is no cord used to cause errors by its stretching. Another type of optical indicator known as Hopkinson's is also used to some extent in England. It differs essentially from the manograph in having a piston instead of a diaphragm. It is not as easily adaptable as the manograph for use on automobile, marine, aeroplane and other engines usually built with an enclosed crank-case as it is intended to be driven by connection to the cross-head or piston.

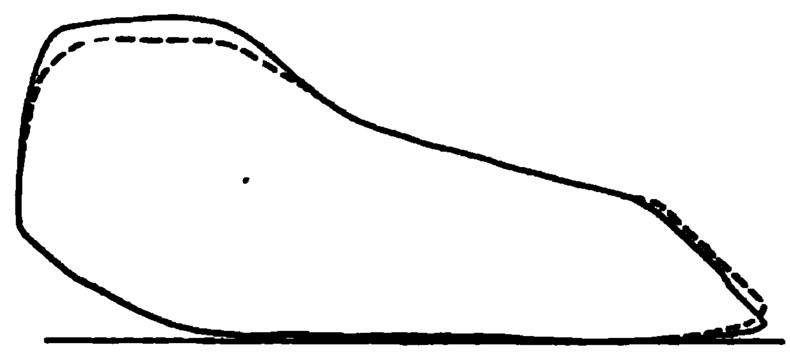


FIG. 164. — Inaccurate Diagram Due to Indicator Cock being partly Closed.

7. **Throttling.** Error in an indicator diagram due to not opening the indicator cock completely is shown in **Fig. 164.** Dotted lines are due to this throttling in the cock.

8. **Errors in the Determination of the Scale or "Number" of Springs** are by far the most important in indicator practice. Power plant engineers rely altogether too much on the calibrations made by the manufacturers of the indicator when it was new. That it is important to calibrate engine indicators with their springs frequently with some reliable and standard apparatus cannot be too strongly stated. It is absolutely essential that this work should be done both before and after every important engine test. There are not very many indicator springs which after considerable use check very closely to the scales for which they were intended. Errors in calculations of indicated horse power are due more often to inaccurate springs than to any other cause. An engineer cannot safely assume that the true scale of the spring in his indicator corresponds at all accurately to the number stamped on it. It is not unusual to find springs in standard makes of indicators in error as much as four or five per cent.

Calculations of the Indicated Horse Power of an engine show usually the power developed on **one side** of the piston, which is commonly stated by the formula,

$$\text{i.h.p.} = \frac{plan}{33{,}000}. \qquad (25)$$

[1] Piston and pencil indicators are made by the H. Maihak Aktiengessellschaft which give accurate diagrams at from 500 to 600 r.p.m.

Where p = mean effective pressure on the piston, pounds per square inch;
l = length of stroke in **feet**;
a = net area of piston in square inches;[1]
n = number of revolutions per minute.

Of the terms of this equation only one, the mean effective pressure, is obtained from the indicator cards.

If we consider now only **one end** of the cylinder, the steam does work on the piston during a "forward" stroke, and, on the other hand, the **piston does work on the steam** on the "return" stroke. Hence to get the mean effective pressure for a stroke the average pressure during the return stroke must be subtracted from the average pressure on the "forward" stroke; and this is obviously the same as the average length of all the ordinates intercepted between the upper and lower lines of the indicator diagram multiplied by the scale of the spring.

Usually the mean effective pressure is found by means of planimeters, the use of which for this purpose was explained on pages 75 to 86. An engineer should, however, know how to calculate the mean effective pressure of an indicator diagram with reasonable accuracy without the use of such instruments. In such cases the **method of ordinates** is very convenient. With suitable draftsman's triangles[2] draw ordinates perpendicular to the atmospheric line at both ends of the diagram as shown in **Fig. 165.**

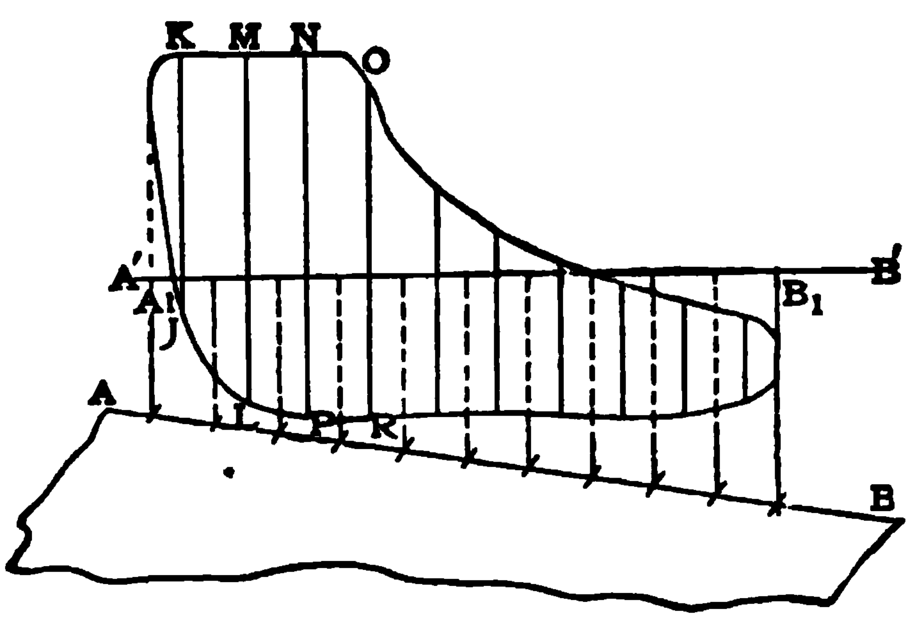


FIG. 165. — Diagram illustrating Method of Mean Ordinates.

Lay off on the edge **AB** of a piece of smooth flat paper, a scale of ten equal divisions so chosen that the total length of the ten divisions is a little greater than the length of any of the indicator diagrams. This scale should then be placed **obliquely** across the diagram to be measured, so that the beginning and end of the scale will be located on the ordinates at the ends of the diagram. Now mark the diagram opposite the divisions of the scale with fine points, and at the **middle** of each of these divisions draw ordinates across the breadth of the diagram. The sum of the lengths of these

[1] In all piston engines the area of the piston rod must be subtracted from the area of the piston on the side where the rod reduces the area effective for the action of the steam or other working substance.

[2] Triangles to be used for this purpose should be tested for accuracy by setting on a straight edge and drawing a vertical line. Then turn over the triangle and observe whether the line drawn coincides with the edge of the triangle.

ordinates divided by ten gives the value of the **mean ordinate**,[1] and this when multiplied by the true scale of the spring gives the **mean effective pressure.** Some time can be saved in summing the ordinates if they are transferred with dividers one after the other to the edge of the strip of paper. The total length laid off divided by ten is then the mean ordinate.

The Engine Constant for Indicated Horse Power. In the use of equation (25), page 141, where

$$\text{i.h.p.} = \frac{\mathbf{plan}}{33{,}000},$$

considerable time can usually be saved when calculating engine tests if the terms

$$\frac{\mathbf{la}}{33{,}000}, \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (26)$$

called the **engine constant** which always remains constant for **each end** of the cylinder, are first computed carefully and then used as constants throughout the calculations. In other words, the **indicated horse power** is found for each end of the cylinder by taking the product of the terms,

$$\textbf{Engine Constant} \times \mathbf{p} \times \mathbf{n}.$$

Indicated Horse Power of Rotary Engines. Fig. 166 shows by a simple diagram a typical rotary engine. The steam inlet is at I and

I E

FIG. 166.—Diagram of Typical "Rotary" Engine.

the exhaust is at **E.** A sliding blade **P,** corresponding to a piston, moves back and forth through the rotor R, as the latter revolves. In the figure **P** is shown at the point of cut-off, the dotted shading indicating the full charge of steam. Equation (25) can be written,[2]

$$\frac{pn}{33{,}000}\left[\text{volume swept through by piston}\left(\frac{\text{cu. in.}}{12}\right)\right]$$

[1] Methods of calculating areas of irregular figures are given on page 74. The area divided by the length gives the mean ordinate.

[2] In equation (25) l is in feet and a in square inches. The volume in cubic inches must therefore be divided by 12 to permit substitution in the equation.

A hole should be tapped through the casing near the top for the attachment of an indicator to determine the mean effective pressure (p) throughout the cycle. This hole can also be used to determine, by filling with water, the maximum volume in the cycle (method explained on page 293), and these data together with the number of revolutions per minute (n) serve for calculating with considerable accuracy the indicated horse power.

The steam consumption per i.h.p. per hour for all types of such engines, if well made and when new, is about 100 to 125 pounds. It is difficult to take up wear in such engines, so that after use for a short time much steam leaks through without doing work.

Speed Counters. Some kind of mechanical counter is ordinarily used for determining the speed of engines and of other machinery with revolving shafts. For the usual services in testing, a hand speed counter

FIG. 169. — Starrett's Differential Speed Counter.

(**Fig. 169**) is considered most reliable.[1] It is generally applicable, inexpensive, and accurate, so that every engineer should have one.

For slow-speed engines some type of fixed counter (**Fig. 170**) is fre-

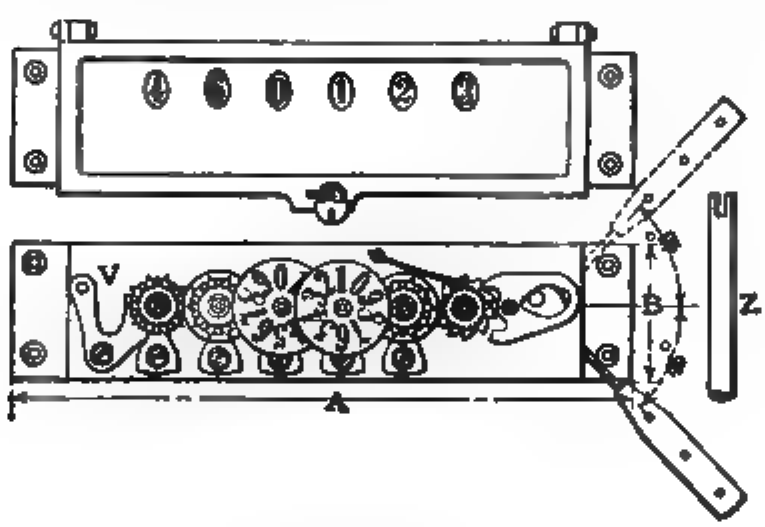


FIG. 170. — Integrating Engine Counter.

quently provided for attachment to the gage board in the engine-room. For gas engines operating by a "hit and miss" method of governing such fixed integrating counters are used in many places for counting the

[1] Starrett's hand counters are generally preferred both in America and abroad.

number of explosions. Actually the number of times the gas valve opens is counted. The greatest trouble with counters of this type is that they will sometimes "stick" even at the normal speeds of stationary engines.

Schaeffer & Budenburg make a pointer and dial **revolution counter** (**Fig.** 171) which is suitable for observing high speeds.

Tachometers, operated centrifugally (**Figs.** 172 and 173) or by the vibrating reed method (**Figs.** 174 and 175) are not accurate enough for the determination of indicated or brake horse power. They can be used conveniently, however, for observing roughly variations in the speed of steam turbines or electric generators when no accurate results are to be calculated from the observations. The vibrating reed type operates by being placed on the frame of the machine and the reed which is most nearly in synchronism with the vibration of the machine indicates by its excessive vibration the speed on a calibrated scale. Belted tachometers because of the added uncertainty regarding the slip of the belt are particularly unreliable.

FIG. 171. — Belt-driven Speed Counter.

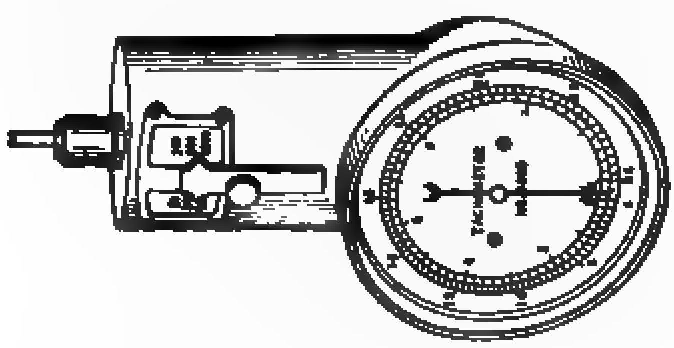


FIG. 172. — Hand Type of Centrifugal Tachometer.

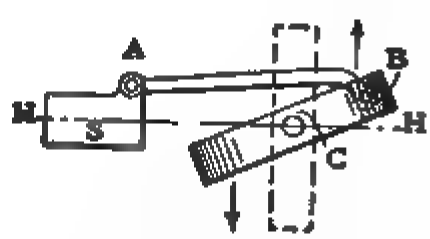


FIG. 173. — Sleeve and Weights in Centrifugal Tachometers.

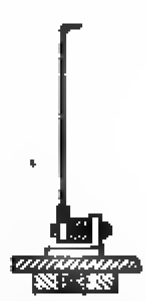


FIG. 174. — Details of Vibrating Reed Tachometer.

Electro-magnetic tachometers depend for their operation on the intensity of the magnetic drag due to flux generated which is proportional to the speed. They are simple in construction, but rather delicate for commercial service and the temperature correction is usually difficult to compensate.

Fluid tachometers are essentially small centrifugal pumps discharging a colored liquid (usually alcohol colored red) into a vertical glass discharge pipe. The blades of the wheel are radial so that the instrument registers the same when running in either direction. The greater the speed the higher the liquid will stand in the tube. Since the height to

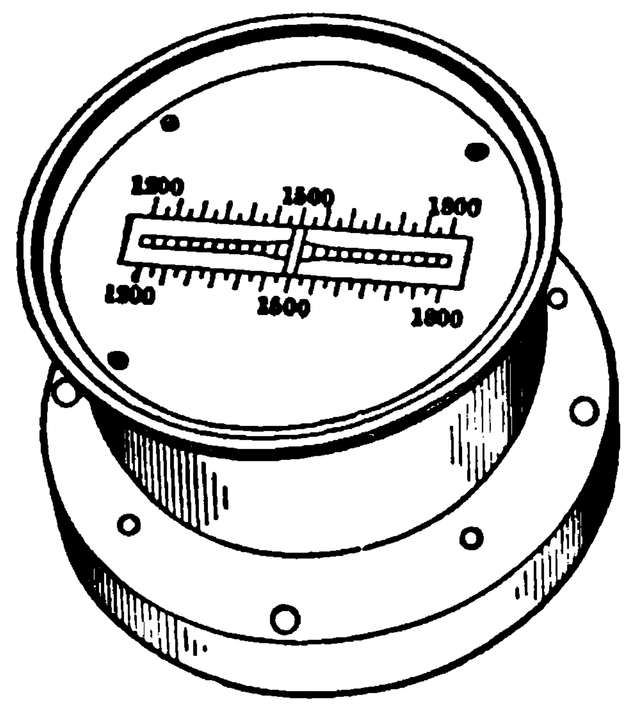


FIG. 175. — Commercial Form of Vibrating Reed Tachometer.

which the liquid will be forced in the tube varies approximately as the square of the speed of the wheel, the upper part of these tubes has a very much more open scale than near the bottom where accurate observations are difficult. Speeds less than 300 or 400 r.p.m. cannot be satisfactorily observed with such instruments.

CHAPTER VI

MEASUREMENT OF POWER — DYNAMOMETERS

A DYNAMOMETER, according to its derivation, is an instrument for measuring force or "power." These are of two kinds:

1. Those **absorbing** the power by friction and dissipating it as heat.
2. Those **transmitting** or passing on the power they measure, thus wasting only a small part in friction.

Absorption Dynamometers (Prony Brakes). Of the class of dynamometers in which the power received is all **absorbed in friction,** the type generally used is called a **Prony brake,** named for Rev. John Prony, who many years ago developed a device of this kind for measuring power.[1] One of the simplest forms is shown in **Fig. 180.**

It consists of a lever **A,** from which a weight **w** is suspended from one end, and a block **B,** supported on a revolving drum or pulley, is attached to the other end. A strap to which wooden cleats are fastened is held in place and tightened by the thumb-nuts **N, N.** When the friction on the strap and block just **balances** the weight **w,** the lever arm **A** is horizontal and the apparatus is in adjustment. Stops, marked **S S,** are provided so as to limit the movement of the lever arm.

When the brake is adjusted or " balanced," the work done in a given time in producing the friction (the power absorbed) is measured by the weight moved multiplied by the distance it would pass through in that time if free to move. Then if,

$\mathbf{r}$ = length of brake arm in **feet.**[2]

$\mathbf{n}$ = revolutions of the shaft per minute.

$\mathbf{w}$ = weight on the brake arm in pounds.

$$\textbf{Brake Horse Power (b.h.p.)} = \frac{2\pi \mathbf{rnw}}{33{,}000}. \qquad (27)$$

[1] Strictly speaking, a brake of this kind does not provide means for directly measuring power, as, for example, horse power, because the element of time is not indicated. In other words, it measures the **tangential force,** of which a couple (**torque**) in linear-weight units, such as foot-pounds, can be computed.

[2] The length of the brake arm is measured by the perpendicular distance from the line of action of the weight **w** to the center of the wheel. When the arm **A is horizontal,** as in **Fig. 180,** the length of the brake arm is usually measured by the horizontal distance from **P** to a line passing through the center of rotation perpendicular to the arm.

In equation (27) the fraction $\frac{2\pi r}{33,000}$ is a constant quantity for a given brake and is called the **brake constant.** When a brake like the one in **Fig.** 180 is used the **effective** weight of the brake itself as weighed at the point **P** must be **added** to the weight **w**.

According to Bach suitable dimensions for a brake of this type are given by $\mathbf{bd} = \frac{12 \text{ b.h.p.}}{\mathbf{k}}$ where **d** is the diameter of the brake pulley in inches, **b** is the breadth of the brake blocks in inches (usually about 1.5 times the diameter of the shaft), **b.h.p.** is the brake horse power to be absorbed, **k** is $\frac{1}{2}$ for air cooling, and varies from 2.5 to 5 for water cooling as the speed increases.[1]

Fig. 180. — Simple Prony Brake.

A very common variation of the Prony brake is illustrated in **Fig.** 181. The block **B** in the preceding figure is replaced by a series of narrow cleats of maple or oak.

Rotation being in the opposite direction from that in **Fig.** 180, the knife-edge at **E** on the arm **A** will now **press** on the pedestal **T**, and the weight **w** can be determined by weighing the pressure on a platform-scales **S.** Since the scales receive not only the pressure due to the force producing friction, but also that due to the weights of the brake and of the pedestal, these weights must be determined and are to be subtracted from all of the readings of the scales to obtain the **net weight w** for substitution in equation (27). Weight of the brake and the pedestal, called the **zero reading,** must be obtained with the brake strap **slack,** so that the block **B** will rest as lightly as possible on the pulley. With small engines this zero reading is obtained most accurately by observing the weights on the scales when the brake pulley is turned around by hand first in one direction and then in the other. In this way we obtain for both the brake and pedestal, with rotation in one direction the weight plus the friction due to their own weight, and with rotation in the other direction the same weight minus the same friction. Half the sum of

[1] For information regarding the designing of Prony brakes for absorbing large powers the reader is referred to *Engine and Boiler Trials*, by R. H. Thurston, pages 260–279.

the two readings is, therefore, the weight corresponding to the pressure on the scales due to gravity alone. With large engines it is sometimes difficult to turn them uniformly by hand, so that the zero reading must be obtained by some other method. This is done usually in practice by placing a very small rod on D (Figs. 180 and 181) ver-

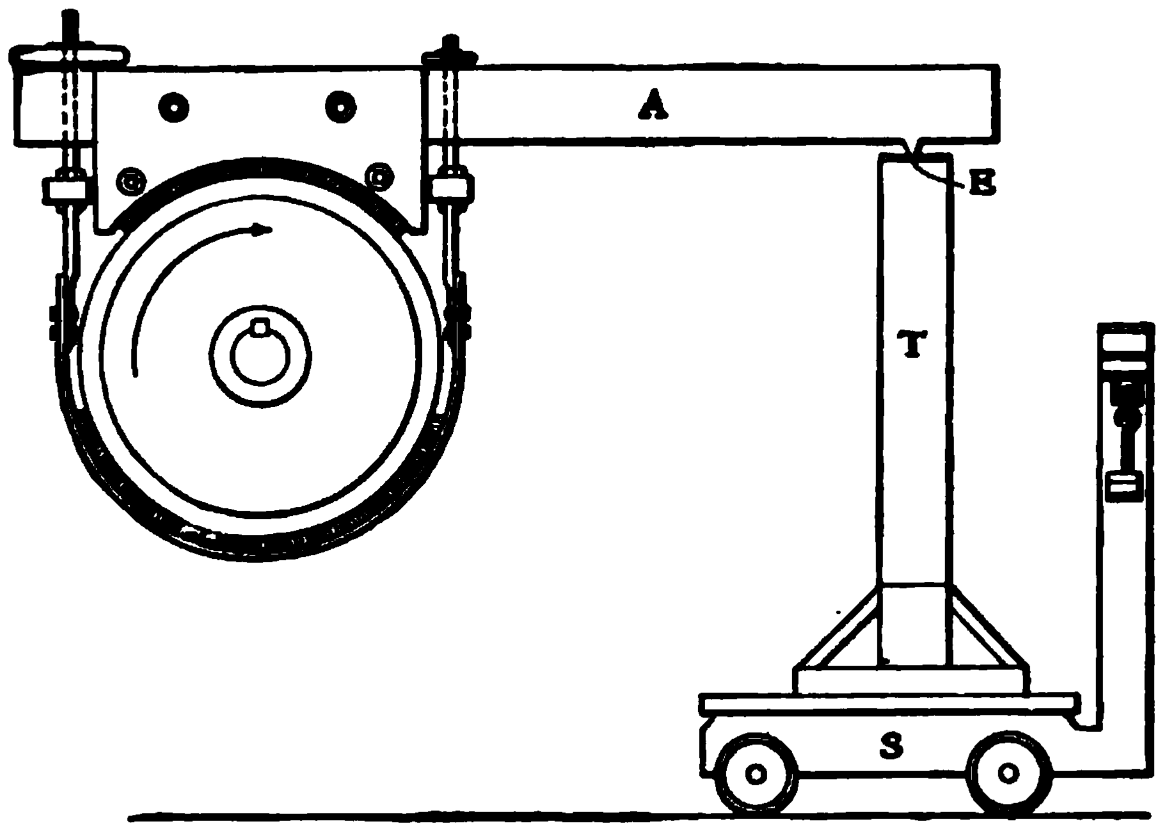


FIG. 181. — Prony Brake with Platform Scales.

tically over the center of the shaft. Then if the strap is loose and due care is observed, the pressure on the scales can be obtained with sufficient accuracy without rotation. The cleats are attached to the bands by wood screws inserted from the outside and countersunk into the bands. Screws used for attaching the cleats to the upper block are inserted through the cleats and countersunk into the wood. At least ½

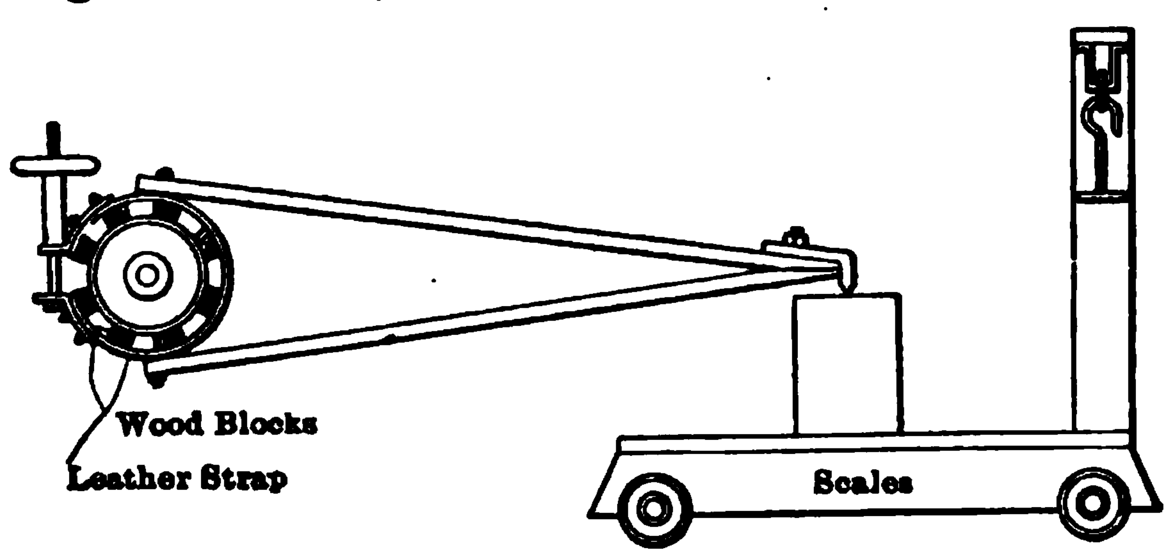


FIG. 182. — "Rough and Ready" Prony Brake.

inch spaces should be left between the cleats to permit air circulation. In all such constructions for dynamometers screws and nails should not touch the rubbing surface as they are likely to cause the friction to be variable and the sound produced is objectionable. Many designers cut grooves into the inside surface of a few of the cleats. These grooves are to be filled with thick grease for lubrication.

A similar arrangement to Fig. 181 is shown in Fig. 182, showing maple

cleats screwed to a leather belt. A piece of old belting is ordinarily used for this purpose so that by this method a very inexpensive Prony brake can be made quickly in any power plant. It has also the important advantage over the two preceding types in that it is readily adjustable to different sizes of pulleys. Washers should be provided for the heads of the screws used to attach the cleats to the belt. If the inside surface of the pulley can be satisfactorily cooled by water the rubbing surface of the cleats need be not more than five square inches per brake horse power at peripheral velocities not over 2,000 feet per minute. For a velocity of 5,000 feet per minute about 10 square inches per brake horse power should be allowed.

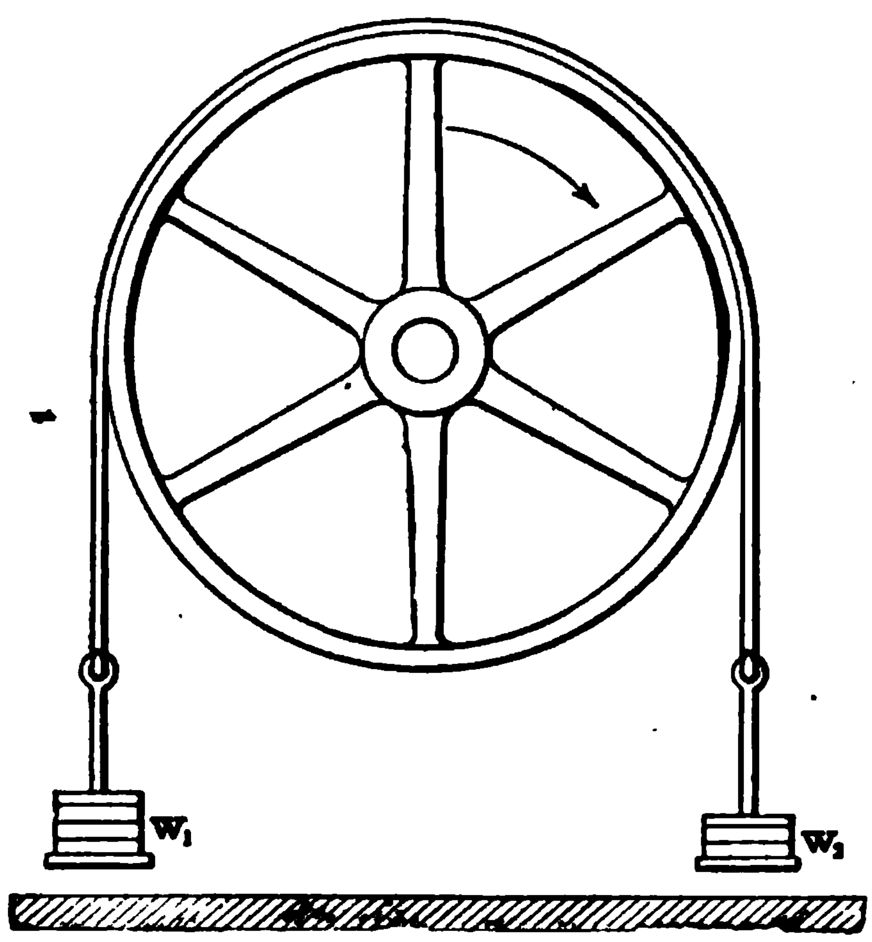


Fig. 183. — Strap Brake.

Another form of Prony brake is illustrated in **Fig. 183**, called a **strap brake.**

It is made up merely of a band of steel or leather or of strands of rope placed over or wrapped around a suitable pulley.[1] In this case weights must be suspended from both sides of the brake wheel or pulley.

In the case of the **strap** brake, **Fig. 183**, the net pull, corresponding to the weight **w**, in equation (27), page 147, is $w_1 - w_2$. Now the same relation would hold if, as is often done, a spring balance is fastened to the floor on say the left-hand side; the pull registered by the spring balance would then be w_1,[2] and the net pull is, as before, $w_1 - w_2$.

[1] It is desirable to use for Pony brakes pulleys of which the section of the face is a double "U," like **Fig. 184.** The outside rims are for keeping the brake in position on the pulley and those on the inside for receiving a small stream of water played upon the inside of the pulley. This stream of water by its evaporation will assist materially to dissipate the heat generated. Such brakes are often operated with pipes arranged to discharge water into the pulley and another pipe to carry it away. This is an excellent system provided the latter pipe is used only to carry away a little overflow, but if so much water is used that there is practically no "steaming," the inside rim of the pulley will fill up with water to be spattered around in every direction as well as over the face of the pulley, where it is particularly objectionable, as it produces variable friction.

[2] To the pull (w_1) must be **added,** however, the weight of any hooks placed between the end of the strap or rope and the spring balance; and if the balance is for any reason suspended in the **inverted** position the weight of the balance itself must also be added.

Brake horse power is calculated then by equation (27), substituting for w the net pull $w_1 - w_2$, so that

$$(\text{b. h. p.}) = \frac{2\,\pi r n\,(w_1 - w_2)}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (28)$$

where r is the radius of the pulley plus half the thickness of the strap, in feet, and n is the number of revolutions per minute. This type of strap brake is very accurate and sensitive, but is suitable only for low powers. About the same friction surface must be allowed as for wooden blocks in Bach's formula.

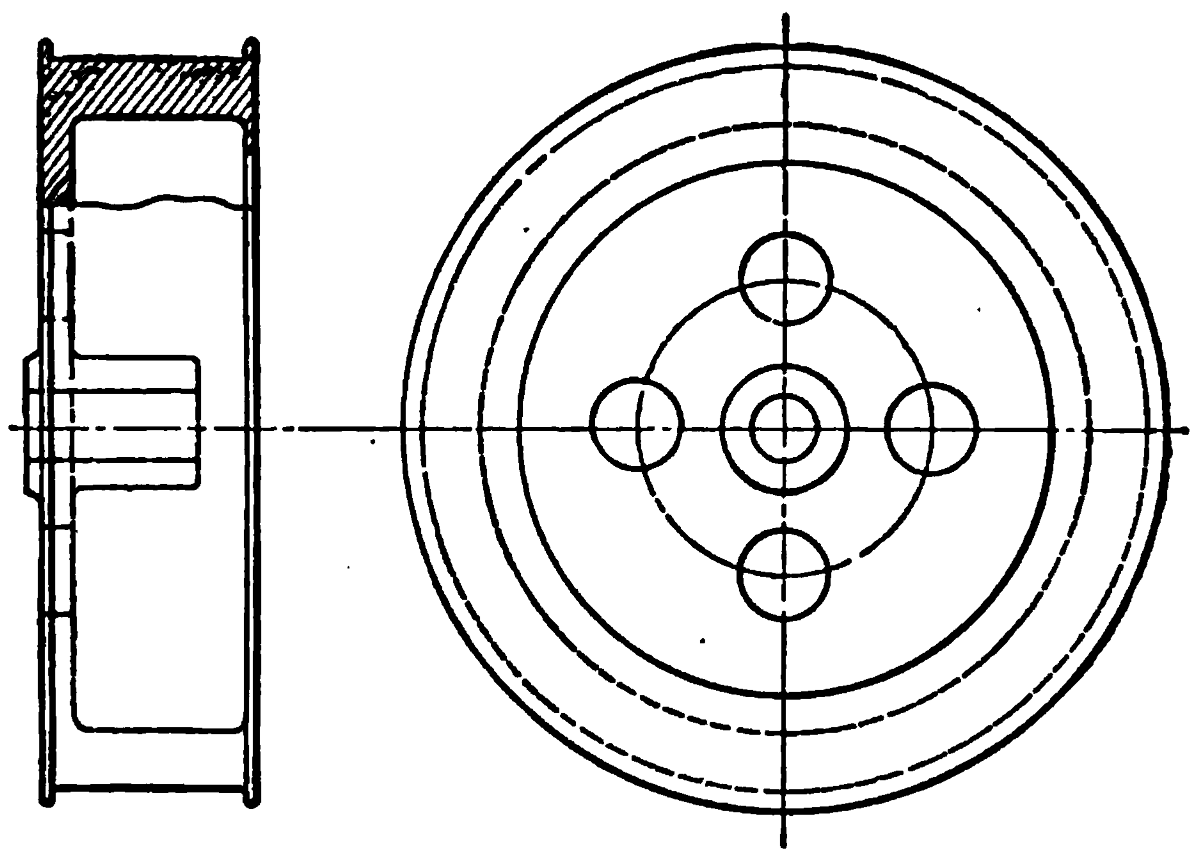


FIG. 184. — Special Pulley for Brakes.

Rope brakes,[1] like the one shown in **Fig. 185**, are much used for "commercial testing" of engines, as they are easily portable or can be made quickly at a small expense from materials always at hand. Moreover, they are self-adjusting, so that accurate fitting is not required. One of this type consists of a rope doubled around a pulley or fly-wheel on the shaft transmitting the power to be measured.

As the friction at the rim of the pulley increases, the tendency will be to lift up the weight w_1. The effect will be to reduce the tension in the end of the rope overhead connected to the spring balance and thus prevent a tendency to further increase of the friction. Several U-shaped distance pieces of wood, preferably maple, are provided to prevent the rope from slipping off the pulley and to keep the parts of the rope separated. These distance pieces should be attached to the rope by soft iron or copper belt lacing, drawn in from the outside of the wooden pieces

[1] Sir William Thomson (Lord Kelvin) invented in 1872 the first rope brake of which we have any record. Although he utilized this device as a friction brake, it was not used by him as a dynamometer.

through the center of the rope, instead of being fastened with screws or nails on the inside, which will heat to a high temperature and then char the rope. Sometimes such fastenings when of hard metal will cut grooves into the surface of the pulley. In this arrangement the spring balance must be supported from some point overhead.

An **anchoring rope** or safety stop securely attached to the weight w_1 should be provided to prevent the weights going over the wrong way

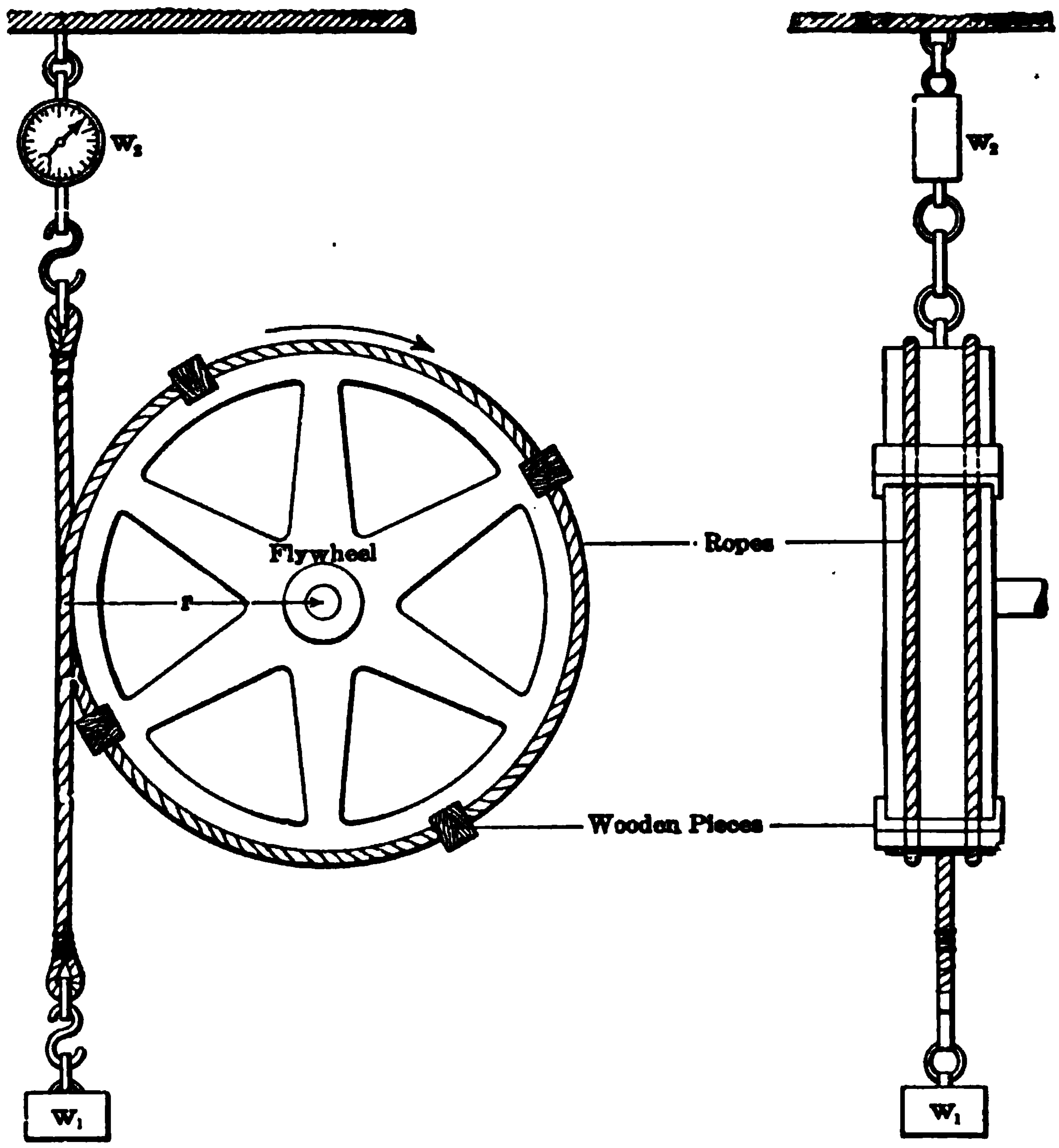


FIG. 185. — Rope Brake.

when starting or stopping an engine in case the valve is not very well set. Its weight or the weight of that part suspended from the weights must be included in w_1. Similarly the weight of the rope between the spring balance and the point where it touches the pulley should be deducted from the readings of the spring balance in accurate work. Spring balances require frequent and very careful calibrations. A modification of practically the same rope brake is shown in **Fig. 186,** where the rope is fastened at the top and bottom to a frame resting on a platform scales.

Equation (28) is used also for calculating the brake horse power for a rope brake, except that r becomes the radius of the pulley or brake wheel plus half the diameter of the rope. All in feet.

Rope brakes are best arranged with the rope placed on the pulley double as in the figures so as to form a loop for supporting the weights. A brake made in this way of double ¾-inch rope and provided with six cleats each of about ten square inches of rubbing surface will absorb fifty horse power if the pulley carrying the brake is about three feet in diameter, and the speed is not over 300 revolutions per minute. For absorbing smaller powers half-inch rope with 4 cleats can be used.

FIG. 186. — Rope Brake with Standard.

Manilla or cotton rope is generally preferred. By steeping the rope in a mixture of deflocculated graphite and melted tallow its frictional properties are improved.

Water-jacketed Bands. A very good method of putting a brake on a wide wooden pulley is shown in **Fig. 187**. Outside of the steel band is a similar band of rubber and canvas. Canvas and steel bands are riveted along the edges, making the space between the canvas and steel a water-tight compartment. Connections are made with the water supply and drain at the ends of the brake strap by means of flexible hose, and a current of water is kept circulating round the wheel, quickly removing the heat generated by friction.

The brake strap may be of almost any width, 20 inches being that used by Professor Goss. The face of the pulley must be **cylindrical and not rounded,** and any inequalities in the face should be made good

before proceeding to use it. Especially is this the case at the joint in split pulleys, and if any space between the halves is left at all, this should be filled with glued wooden plugs. The thickness of a steel band used on a pulley 24 inches in diameter was No. 12 gage. A layer of rubber should be inserted at the joints between the canvas and steel band, so as to ensure a good joint, and special cast-iron ends are usually riveted to the brakestrap and fitted with water connections.

FIG. 187. — Water-jacketed Band for Brake.

A somewhat similar brake has been used at the Pennsylvania State College, consisting of a flat strap with both sides made of copper. It is applied to the fly-wheel, much in the same manner as the Prony brake. The ends are coupled together by an adjustable screw. The pressure of the brake strap on the wheel is produced and regulated by water flowing through the tube, much in the same way as in the Alden dynamometer (page 155). If the wheel surface is maintained in a well-lubricated condition, the wear of the copper tube is inappreciable. In a modified form of this brake a thin metallic band is interposed between the copper tube and the wheel, and consequently no wear of the tube takes place.

Fan Brakes or Dynamometers. For determining the power of high-speed engines a dynamometer consisting of two flat fan blades arranged

FIG. 188. — Fan Brake.

to be attached to the engine shaft is very convenient. Power is absorbed by the "fan" action of the plates on the surrounding air.

Amount of power absorbed by fan blades depends (1) on the size of the blades, (2) their distance from the center of rotation, and (3) on approximately the cube of the number of revolutions per minute. **Fig. 188** shows a somewhat elaborate testing-base for automobile and marine engines. The engine is connected by means of a universal coupling **C** to the shaft **S** to which the fan blades **E** are attached. The frame **Q** is for supporting the engine. A tachometer **J** is used to indicate the approximate speed. The fan is shown enclosed in a glass frame-work **M** to prevent the air currents from other engines, etc., from interference with the discharge of air from the fan blades. **Fig. 189** shows typical calibration curves for the fan in **Fig. 188.** Fans for this purpose are usually calibrated by attaching them to a variable-speed electric motor of which the efficiency curve is known. For appropriate work a fan built as shown in the figure can be used for testing and even without the casing **M** the curves will show satisfactory values of power with a probable error of less than three per cent. The blades are ten inches wide in the radial direction, fourteen inches wide in the axial direction, and ⅛ inch thick. Numbers on the curves indicate inches from the center of the shaft to the middle of the blades. This distance can be varied by shifting the bolts. The speed to be used in accurate calculations should be taken on a high-speed engine with a good hand-counter.

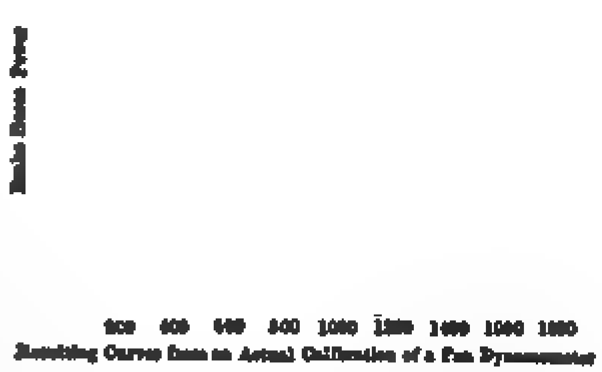


FIG. 189. — Curves for Fan Brake in Fig. 188, with 10″ × 14″ Blades.

Alden Brake. An entirely different type of absorption dynamometer is the Alden brake, illustrated in **Fig. 190.**

In this apparatus the rubbing surfaces producing the friction necessary for absorbing the power are separated by a film of oil, and the heat generated is carried off by a stream of water circulating as indicated by arrows. It consists of a disk of cast iron **A**, which is connected to the shaft **S**, transmitting the power. This disk revolves between two thin copper plates **E, E**, fastened together at their outer edges to form a shallow cylinder which is filled with a bath of heavy (cylinder) oil. Water under pressure is discharged into the chamber adjoining the copper plates and any increase in pressure causes the copper plates to press toward the cast-iron plate **A** with more force. The friction of the thin film of oil between these copper and iron plates tends to turn those of copper, but as they are rigidly connected to the outside casing **C** carrying the brake arm **P**, the tendency to turn can be determined by

weighing as with a Prony brake. To maintain the moment of resistance constant under all circumstances the pressure and consequently the flow of water into the casing is automatically regulated by a cylindrical valve V, which becomes partially closed if the brake arm moves above a certain horizontal position. This valve is shown in section in Fig. 191. The end at W is connected to the water main and the other end Y to the brake casing by means of a right-angled bend R (Fig. 190). Water entering by the pipe W passes through the ports N and then through the ports H into the pipe Y. Now a small angular movement of the pipe

Fig. 190.—Alden Brake.

W, relative to the pipe Y, will open or close the ports H and thus regulate the supply of water. These ports are very narrow, so that a very small angular motion is sufficient to close them. By making the outer casing of the valve of rubber it is found to be sufficiently flexible to permit moving the valves and at the same time it offers very little resistance to the movement of the casing.

Reynolds-Froude Dynamometer. Professor Osborn Reynolds has designed a modification of the Froude brake which is shown in section in Figs. 192 and 193. This device has a very large capacity for absorbing power. One of these brakes with a single rotor only 30 inches in diameter will absorb 700 horse power at 200 revolutions per minute.

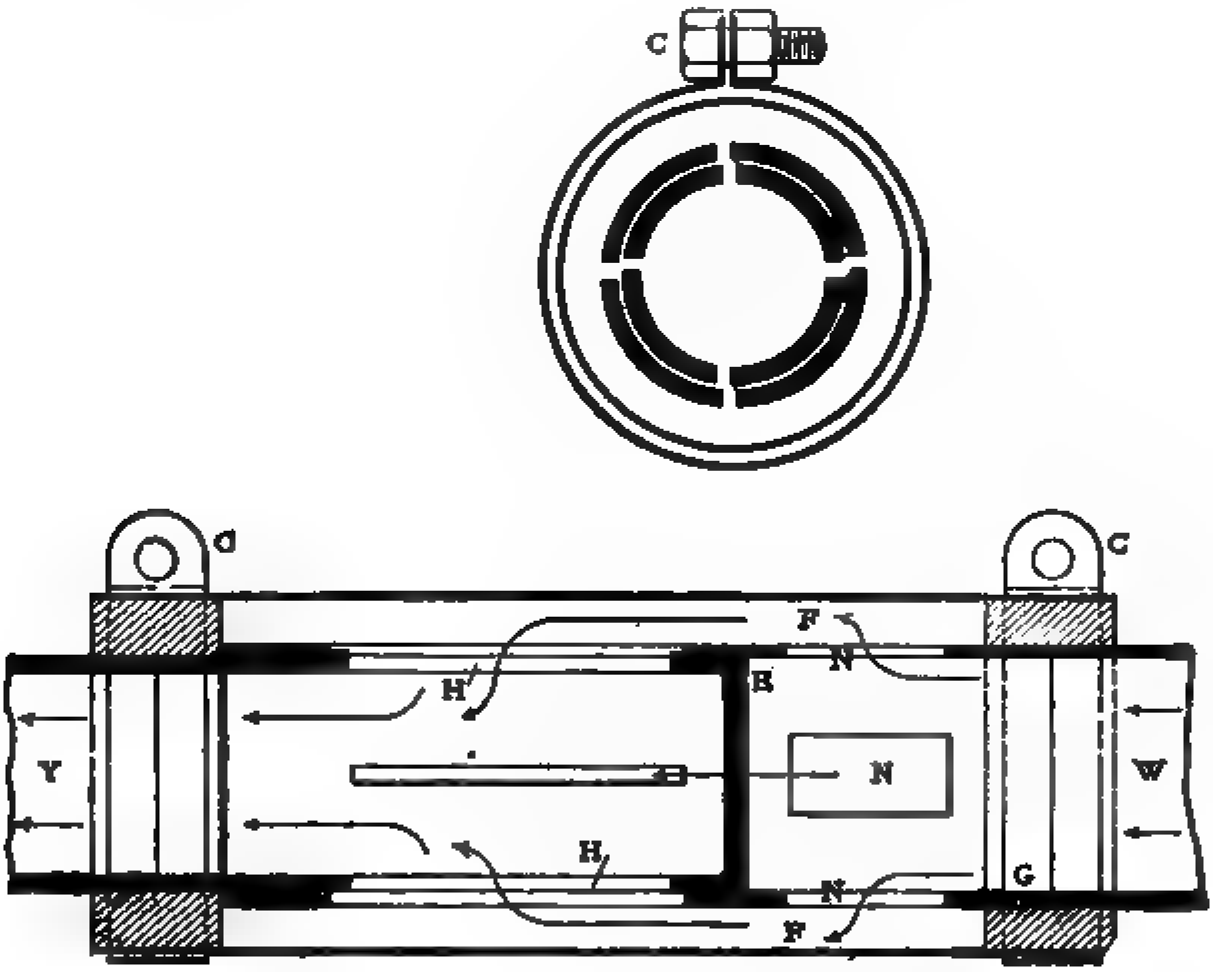


Fig. 191. — Regulating Valve in Alden Brake.

Fig. 192. — Reynolds-Froude Dynamometer.

Fig. 193. — Reynolds-Froude Dynamometer Shown Diagrammatically.

Water is supplied to the apparatus through the flexible pipe marked **E** (a rubber hose is very satisfactory) from which it passes in the direction of the arrows into the space **H**, (**Fig.** 193), then into the centers of the vortices through the holes **J** (shown dotted) which are drilled

through the walls of the pockets. From these pockets the water passes between the rotor and the casing into the space **M** from which it discharges through the discharge outlet **D** into the drain.

If any air collects in the center of the vortices it can escape through the holes **I** in the pocket walls in the casing into the cored channel **L** and the air outlet pipe **O.** If any water comes through these passages with the air it is carried off through a funnel and pipe emptying into the main discharge pipe **D.** The brake horse power (**b.h.p.**) for this dynamometer is expressed on a very conservative rating by the equation

$$\text{b.h.p.} = \frac{3.18}{10^{10}} n^3 d^5,$$

where **n** is the number of revolutions per minute and **d** is the diameter of the rotor in inches.

Water Brakes. Power can also be absorbed by moving in water a rotor similar to those in steam turbines. Such an apparatus is called a turbine water brake. A good example is shown in **Figs. 194** and **195.**

FIG. 194. — Westinghouse Water Brake.

It is the type used by the Westinghouse Machine Company of Pittsburg for testing the power of large steam turbines. It consists of a rotor **R,** mounted on a shaft **S, S,** of which one end is arranged to be connected directly by means of a coupling to the shaft of the turbine or engine to be tested. This rotor revolves within a closed casing **Z,** supported on the journals **J, J,** through which the shaft passes. Around the periphery of the rotor there is a series of rows of vanes which when revolving tend

to give to any water contained in the casing a rotary motion. Between every two rows of vanes on the rotor there is fitted a row of stationary vanes attached to the inside of the casing. This arrangement is illustrated diagrammatically in **Fig. 195**, where the cross-hatched sections represent the stationary vanes and the "solid" sections the moving vanes. The tendency of the moving vanes to produce this rotary motion of the water is, as it were, resisted by the stationary ones, and this action develops a very large amount of heating, due to fluid friction in a manner analogous to the operation of a Prony brake or any other absorption dynamometer. As a result of this friction a force is developed tending to turn the casing in the direction of motion of the rotor. The casing, however, is prevented from turning by a radial arm bearing down on a platform scales. The intensity of this tendency to turn can be regulated as in the Alden brake (page 155) by adjusting the valves controlling the flow of water through the casing. The vanes on the rotor are of the

FIG. 195. — Vanes of Westinghouse Water Brake.

same kind as used in steam turbines and have the function of imparting a high velocity to the water flowing through them in an axial direction. Water enters the casing through the inlet pipes **C, C′** (**Fig. 194**), discharging a stream from both sides of the casing toward the central portion of the rotor. At the middle of the periphery of the rotor there are a number of slots or ports **A, A, A, A** through which the water discharges to the right and left, passing first through a broad central row of stationary vanes shown in **Fig. 195**. Escaping from these vanes it is picked up by the rows of moving vanes on each side, which give it a high velocity and, in turn, discharge it into the adjacent rows of stationary vanes where the velocity just acquired is checked, and so on across the face of the rotor, the moving vanes adding velocity only to be lost in the next row of stationary vanes. From the last rows of moving vanes the water is discharged into semicircular passages **B, B′**, which direct the flow of water into the center of the rotor when the cycle is repeated. The water brake illustrated here was designed for high powers, so that a number of rows of vanes was necessary. For small powers of, for example, from 200 to

500 horse power, not more than two rows of moving vanes with the corresponding number of stationary vanes would be required.[1]

Power absorbed by a water brake is calculated in the same way as for an ordinary Prony or rope brake.

In the operation of the brake the water is quickly raised to the boiling-point and a considerable portion evaporates, carrying off as steam very large quantities of heat. Vents for the escape of this steam are indicated by **D** and **D′** in the figure, showing the cross-section of the brake. Unless considerably more water, however, was admitted to the casing than was required to replace that lost by evaporation, the action of the brake would be more or less irregular, so that an excess of water is supplied and there is a constant discharge of hot water through the passages marked **E** and **E′**.

D D

Fig. 196. — Simple Water Brake.

Another type of water brake is shown in Figs. 196 and 197 which is also used for testing engines and turbines.[2] The revolving paddle wheel or "runner" **R** is designed to run in very close clearance with the serrated rim piece **P**. The outer casing is rigidly attached to the lever arm **A** made of such a length as to facilitate rapid calculation. A little roller or wheel **W** on the end of this arm rests upon the platform scales used to weigh the load.

Water is admitted through a flexible hose connection at the opening **I** and from there enters the interior of the wheel. When the wheel is revolving this water is thrown out by centrifugal force through small holes drilled in the rim, entering into the outer teeth spaces. In this passage of the

Fig. 197. — Section of Brake shown in Fig. 196.

[1] To calculate the resistance of such vanes see "The Steam Turbine," by the author, pages 115–125.

[2] A very satisfactory water brake is also made by the Michigan Motor Specialties Co., Woodbridge Street, Detroit. It has an extensive sale for automobile testing.

water a considerable fluid resistance is produced. The water finally escapes after passing through this tortuous passage through the close clearance around the outside of the wheel and discharges through the pipes connected at **D**.

Webb's "Viscous" Dynamometer, Fig. 198, consists of a number of flat steel plates P_1 (circular saw blanks are excellent) fastened to a hub **H** which is keyed to the driving shaft **S**. Between these plates on the shaft are other plates P_2 rigidly attached to the casing **C**. When a liquid is put into the casing the rotating plates P_1 tend to carry it around with them by the action of viscosity, and the turning moment of the shaft **S** is communicated to the fixed plates P_2 and to the casing. The casing is supported on pedestals **E** provided with ball-bearings. The turning moment on the casing is balanced by weights placed on a horizontal arm as arranged for the Alden dynamometer (Fig. 190) or may be set up so as to press on a platform scale as in Fig. 181. The latter method is probably the better. Water is most commonly used as the liquid. It enters through the funnel **F** near the center of the casing and leaves at the discharge pipe **D**. A flow of liquid is maintained which carries away the heat generated. The power absorbed varies as the cube of the speed and the fifth power of the diameter. The frictional or "viscous" resistance can therefore be varied by adjusting the depth of the water in the casing. Usually a number of holes are made in both sets of plates through which water can pass to the discharge connection without having the casing filled to the tips of the fixed disks before overflowing to the next compartment. The quantity of water in the casing can be regulated by both the inlet and discharge valves. It is generally best to adjust both valves at the same time for large variations in load. Viscosity of the water decreases somewhat with rise of temperature of the water.

FIG. 198. — Webb's "Viscous" Dynamometer.

Professor Webb has also arranged in some of his designs to supply the water through a hollow shaft and regulate the supply by a piston valve to be adjusted axially to supply a varying number of compartments.

Regulation is accomplished not only by varying the radial depth of the water but also by changing the number of compartments containing water. In this latter arrangement no holes are required in the fixed or stationary plates to permit draining. Discharge connections are provided for each compartment so that there are as many drain pipes as compartments.

The dynamometer starts easily and without load so that it is better suited for use on steam turbines than those of the Alden and Froude types. The steel plates are the only parts subjected to high centrifugal stress and they are strong enough for all practicable speeds.

This apparatus is not large for the power absorbed. One of these dynamometers provided with two disks, each two feet in diameter, absorbed 180 brake horse power when running at 2500 revolutions per minute with a radial depth of three inches of water, and with one inch of radial depth of water 60 horse power can be absorbed. Brake horse power (b.h.p.) absorbed by each rotating plate is approximately

$$\text{b.h.p.} = \frac{n^3(r_2^5 - r_1^5)}{130{,}000{,}000},$$

where n is the revolutions per minute, r_2 is the radius of the moving plate in feet and r_1 is the inner radius of the annular ring of water in the casing.

Dynamos (Electric Generators and Motors) as Power Dynamometers. One of the most convenient means for measuring the power of high-speed engines and turbines is to connect an electric generator to the main shaft as in **Fig. 199.** Then if the efficiency of the generator is known at the particular speed and output at which it is to be operated, a very accurate method of measuring the power of the engine or of any other type of motor becomes readily available. The output of the generator should be determined by observations of the volts and amperes with carefully calibrated portable instruments. Remembering that for direct-current generators volts times amperes gives watts and that 746 watts are equivalent to one horse power, then if e.h.p. is the horsepower output of the generator we have

FIG. 199. — Electric Generator used as Power Dynamometer.

$$\text{e.h.p.} = \frac{\text{volts} \times \text{amperes}}{746}.$$

It is not unusual to hear this result called the "**Electrical**" **horse power** of the engine or turbine.[1]

The actual horse power delivered to the generator is, of course, the **brake horse power**, and into this result the efficiency of the generator enters. Thus, for direct- (or continuous) current generators

$$\text{b.h.p.} = \frac{\text{volts} \times \text{amperes}}{746 \times \text{efficiency of generator}}.$$

The load on the generator should be maintained uniform by absorbing the electrical output in lamp or wire resistances for small powers but for larger powers a water resistance or rheostat is generally used. Electrodes are generally made of ¼- to ½-inch iron or steel plates, allowing about 1 square inch per ampere.

As a rule electric **motors** are very serviceable in mechanical engineering laboratories as power dynamometers. The efficiency is easily obtained, the usual method being to determine an efficiency curve for varying power inputs by a Prony brake test. This efficiency is

$$\text{Efficiency of motor} = \frac{\text{b.h.p.}}{\text{e.h.p.}},$$

where **e.h.p.** is the "electrical" horse power input, as measured with voltmeters and ammeters.

If then a pump, air compressor, ventilating fan or a similar machine is to be tested by the electrical method, it should be direct-connected to the shaft of the motor, and its efficiency **E** will be

$$E = \frac{\text{u.h.p.}}{\text{e.h.p.} \times \text{efficiency of motor}},$$

where **u.h.p.** is the useful work done by the machine, in horse power.

This last method serves also as a convenient method for obtaining the efficiency of generators, since by connecting it directly to the shaft of a motor previously "calibrated" (for efficiency) the electrical output of the generator and the input to the motor are readily determined.

When a so-called **variable-speed** motor is used as a dynamometer, its efficiency must be determined at the particular speed and power at which it will operate when driving the machine to be tested.

Eddy-current Brakes[2] are built with a number of electro-magnets and one or more copper disks. Either the coils or the copper disks may be

[1] When an alternating-current generator is used the power factor must also be measured with a suitable instrument. In this case, the actual e.h.p. is (volts × amperes × power factor) ÷ 746.

[2] For more information regarding the design of this type see "Eddy-current Brakes" in *Journ. Inst. E. E.*, (London) vol. 35, (1904–5).

rotated with the shaft while the other is held stationary. Eddy currents generated by rotation with the electro-magnets excited produce a resistance of which the moment is measured by a lever arm and scales as with similar forms of dynamometers.

Transmission Dynamometers. Instruments of this type are used to measure the amount of power transmitted without absorbing any more power than is absolutely needed to move the dynamometer.

Goss Belt Dynamometer. One of the simplest forms of transmission dynamometers, designed by Professor W. F. M. Goss, is illustrated in **Fig. 200** by a line drawing. Being so simple in construction, it can readily be made in any factory or workshop with the materials available. It is essentially a **differential lever** measuring the difference in tension between the two sides of a belt. This lever is pivoted at the point **D** and to it are attached the shafts carrying the pulleys **A** and **B**. Weight hangers are attached to the ends of the beam. The beam and hanger must be balanced to be in the horizontal position, that is, in the position of equilibrium when the belt is not moving. Power transmitted is measured by the product of the speed of the belt and the difference in belt tension between the two sides of the dynamometer.

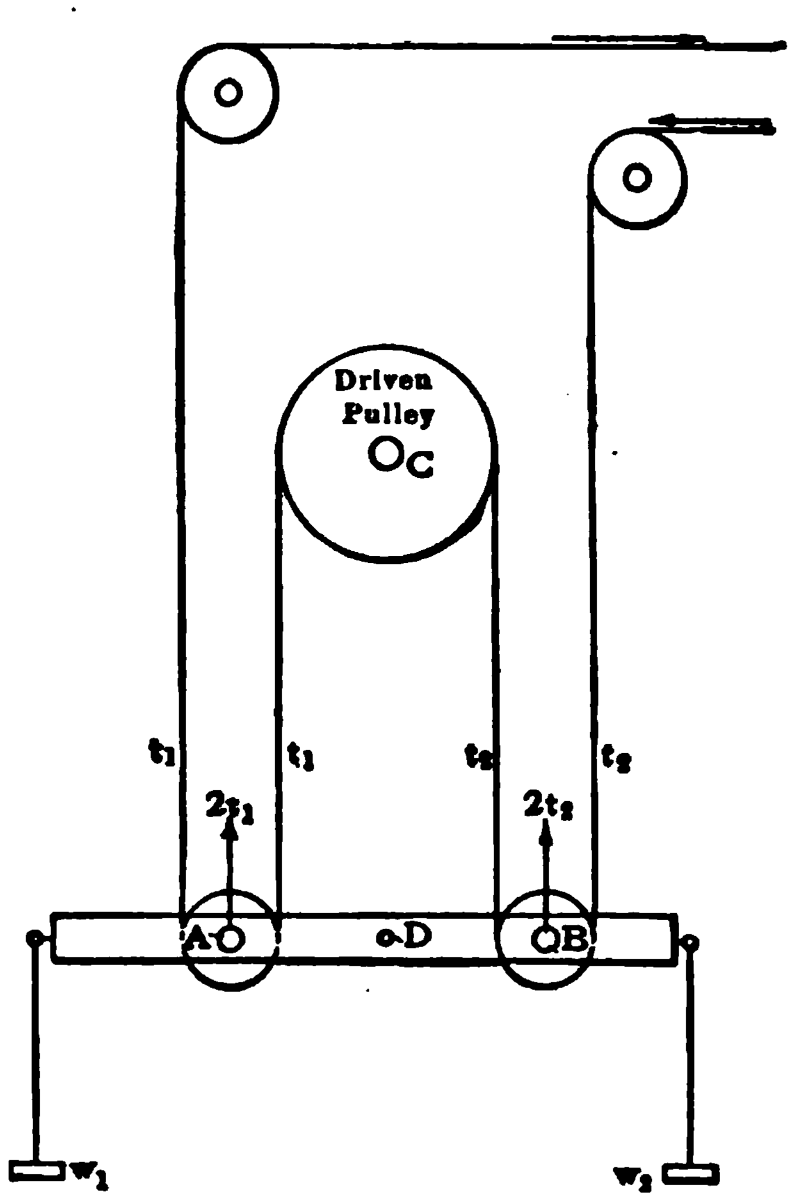


Fig. 200. — Goss Belt Dynamometer.

The force tending to raise the left-hand end of the lever is to be twice the tension t_1 of the tight side of the belt, while that raising the right-hand side is twice the tension t_2 in the slack side of the belt. These tensions on the two sides of the lever can be measured by weights w_1 and w_2 suspended from the hangers. The force tending to rotate the lever is therefore twice the difference in tension on the two sides of the belt $2\ (t_1 - t_2)$ and this force acts with a leverage $AD = DB = r$, when these two arms are made equal. If now the distance from the pivot **D** to the point of support of the weight w_1 is made twice $AD = 2\ r$, and if the weight w_1 is made equal to the difference in the tensions it will balance the lever. For these conditions, taking moments about **D**,

$$w_1 \times 2\ r - 2\ t_1 r + 2\ t_2 r = 0,$$
$$2\ w_1 = 2\ (t_1 - t_2),$$
$$w_1 = t_1 - t_2.$$

Now in any transmission system the difference in the tension of a belt or a rope on the two sides of a driven pulley multiplied by its speed is a measure of its power. If, then, **d** is the diameter of the driven pulley **C**, plus the thickness of the belt or rope in feet, **n** is the number of revolutions per minute of this pulley, and $\mathbf{w}_1$ is the weight in pounds on the left-hand side required to **balance** the lever; when power is transmitted, then,

$$\textbf{Horse power transmitted} = \frac{\pi d n w_1}{33{,}000}. \quad . \quad . \quad . \quad . \quad (29)$$

To reduce the vibrations of the apparatus a dash-pot is connected to the right-hand side. To prevent excessive movement of the lever when unbalanced, stops are placed above and below the lever on the left-hand side.

Another form much used for mill purposes and fan testing is the belt dynamometer, shown in **Fig. 201.** The driving pulley is **A** and the driven pulley is **B.** The connecting belt passes over the pulleys **C** and **D.**

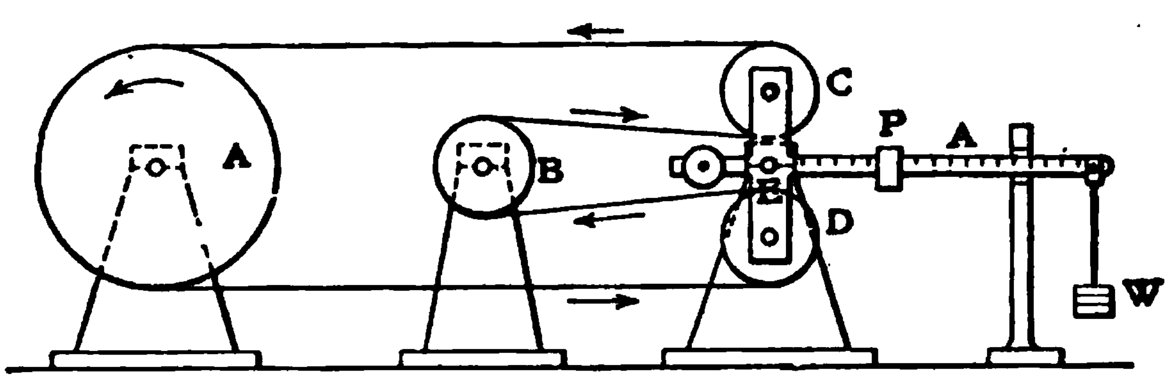


Fig. 201. — Compact Belt Dynamometer.

These two pulleys are mounted on the frame of the dynamometer carrying a scale beam **A,** all of which turns on the center of support. The difference in the total stress on the two sides of the belt is computed by multiplying the net weight on the beam by the equivalent leverage, the latter being found by dividing the length of the beam by the distance from the center of support to the centers of pulleys **C** or **D.**

Differential Dynamometers. The apparatus illustrated in **Fig. 202** is typical of a number of dynamometers indicating by means of a differential lever operated by gearing, the amount of power transmitted.[1] This is a very common form of **transmission dynamometer.** Power is received from the motor (or engine) by the shaft **A,** which is connected only indirectly by means of gears to the shaft **A′** opposite, which transmits the power to the work. To the adjoining ends of these shafts bevel wheels **B** and **D** are attached. The lever **L** turns on an axis **concentric** with the shafts **A** and **A′**, in a plane perpendicular to them. It carries

[1] Similar forms of differential dynamometers are known as White's, King's and Bachelder's. The first instrument of this kind, it is stated, was invented by Samuel White in 1780. These dynamometers are sometimes called **epicyclic,** signifying "wheels traveling around a circle or around another wheel."

bevel wheels **C** and **C_1**, gearing with **B** and **D**, through which the power is transmitted. There is a tendency then for the left-hand end of **L** to go downward and for the right-hand end to rise. If, furthermore, the lever **L** were permitted to revolve, no work would be transmitted from **B** to **D**, and therefore **D** would remain stationary. As these gears are usually proportioned so that **B** revolves with half as many revolutions in a given time as **L**, a force applied at **B** at a given radius from the center will balance a weight twice as large at the same radius on the lever **L**.

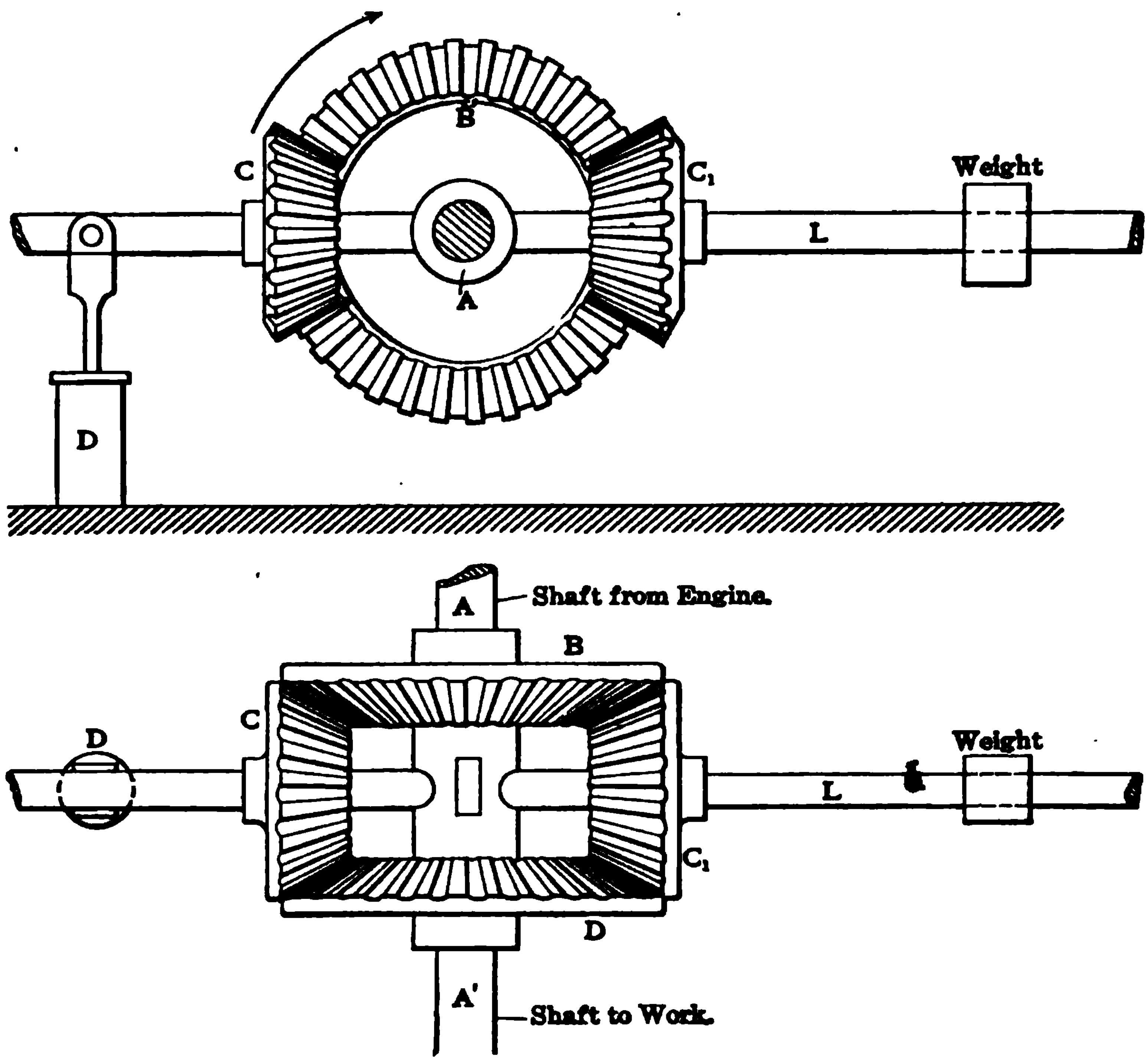


FIG. 202. — Typical Differential Lever Dynamometer.

Then the moment of the force applied to the lever **L** to balance it must be twice as great as the moment of the force transmitted from either **C** or **C_1** to **D**. That is, the sum of the moments applied to **C** and **C_1** must equal the moment applied to **B**. If, therefore, **l** is the length (feet) of the arm at which the weight **w** (pounds) is applied, and **n** is the number of revolutions per minute of the shaft **A**, then the POWER TRANSMITTED PER MINUTE =

$$\frac{2\,\pi lnw}{2}\ \text{(foot-pounds)}, \quad \ldots\ldots \quad (30)$$

and

$$\text{Horse power (h.p.)} = \frac{\pi lnw}{33{,}000}. \quad \ldots\ldots \quad (31)$$

Now if the lever L is to be prevented from turning about its axis, **a couple will be required which is twice the driving couple being transmitted.** If, then, a weight of w pounds sliding on L as shown in the figure is placed at a distance l feet from the axis, so that the lever remains horizontal, the driving couple can be determined; and when the revolutions of the shaft **A** are known, the power can be found. A dash-pot **D** is usually attached to the differential lever L to reduce vibrations. This dash-pot should always be kept filled with glycerine or good clean oil. If the dash-pot is sticky consistent results cannot be expected.

A Webber Differential Transmission Dynamometer as made commercially is illustrated in **Fig. 203.** The scale on the lever arm of this instru-

Fig. 203. — Webber Differential Transmission Dynamometer.

ment is graduated into 100 divisions and a bell is provided which rings at every 100 revolutions. Since the horse power transmitted in one revolution per minute is $\frac{\pi \mathrm{lnw}}{33{,}000}$, equation (31), then the horse power corresponding to one division on the scale per 100 revolutions per minute is also $\frac{\pi \mathrm{lw}}{33{,}000}$ for a perfect calibration.

It is interesting to observe that if we let

v = the vertical force acting at C and C_1;
p = the vertical pressure between the teeth at each point of contact;
d = distance from the center of the rotation to C and C_1;
l = the distance from the same center to the weight w.

Then from the foregoing discussion it should be clear that $2v = 4p$ and $wl = 2vd = 4pd$. If r is the effective pitch radius of the driving gear wheel **B**, r_1 is the radius of the small bevel wheels, and the force producing the turning movement in the shaft **A** is represented by **f**, we have,

$$fr = 2pr_1,$$

and

$$f = \frac{2pr_1}{r} = \frac{wlr_1}{2dr}. \qquad (32)$$

If we know the number of revolutions, then the space passed through by the force **f** can be calculated, and the work in foot-pounds is the product of the force times the distance passed through. The units given above are of course respectively in feet and pounds.

About ten horse power can be transmitted and measured by one of these instruments having wheels **B** and **D** about ten inches in diameter, when operating at 800 revolutions per minute.[1] Wear on the gears and noise becomes excessive at higher power and speed.

Calibration of a Differential Dynamometer. 1. Examine the dashpot and observe whether the piston moves freely in the cylinder, particularly without "sticking." After the apparatus has been well oiled the position of the poise to make the lever arm horizontal should be observed for "no load." If this is not at zero then all the readings on the scale must be corrected by the amount of this zero reading.

2. At each of the speeds required make a preliminary run without load and observe the reading of the poise when the lever is balanced.

3. Attach a Prony brake to the shaft from which the power is to be transmitted and observe for a series of loads and speeds the readings of the poise on the dynamometer lever.

4. For each speed plot a curve with theoretical **foot-pounds per minute** by equation (30) as abscissas and actual foot-pounds per minute as determined by the Prony brake as ordinates.

Emerson Power Scales. Another very satisfactory instrument for the measuring of power transmitted by shafting is known as the Emerson Power Scales. It is illustrated in **Fig. 204**.

It consists of a pulley **C** keyed to the shaft. To this pulley **C** a wheel **B** is connected loosely **by studs EE projecting between and bearing against its spokes.** The pressure exerted on these studs is proportional

[1] S. S. and W. O. Webber, *Trans. Am. Soc. M. E.*, vol. 4, page 227.

to the power transmitted by means of the pulley **C**, and this pressure is transmitted by a system of levers **LL** and bell-cranks **MM** to a sleeve **A** connected to a "weighing lever" **W**. The sleeve slides loosely on the main shaft. The amount of the pressure exerted on the studs is indicated for small values by a pointer **P**, moving over a graduated scale **F**. For pressures beyond the limits of the graduated scale weights are placed on the scale pan **N**. A dash-pot **D** is provided to prevent excessive vibrations and make the pointer "dead-beat."

The scale **F** is calibrated to read the pressure (**force**) exerted by the torque of the pulleys **C** on the studs **E** in pounds. The work or "power" is calculated, therefore, by taking the product of this force times the distance moved through. If **d** is the diametral distance between the centers of the studs in feet, **n** the revolutions per minute and **w** the reading of the scales, then,

FIG. 204. — Emerson Power Scales.

$$\textbf{Horse power}^{1} = \frac{\pi dnw}{33{,}000}. \quad \quad (33)$$

A speed counter is attached to the apparatus for counting the number of revolutions. This apparatus is made by the Florence Machine Co., Florence, Mass.

Flather's Hydraulic Transmission Dynamometers. A form of transmission dynamometer which is operated by **hydraulic pressure** is shown in **Fig. 205**. The power shaft is keyed to the boss of a pulley **B** with two or more arms carrying hydraulic cylinders **R**. Projecting ends or studs from these cylinders bear upon the arms of a loose pulley **A** on the same shaft. The torque imparted by the driving belt to the loose pulley **A** is thus transmitted to the shaft **S** through the liquid, and the resulting pressure is conveyed by radial pipes **U** to the hollow central shaft, and then to a pressure gage **G**. The hollow shaft is always filled

[1] Compare with (31) for differential dynamometers, page 166.

with oil. In the figure an engine indicator I is shown attached to the hollow shaft for recording the pressure. The loose pulley A drives the tight pulley B through its pistons which press on the oil in the cylinders carried by the tight pulley. By means of a worm drive the drum of the indicator receives its motion from the central shaft S. Figs. 206 and 207 show more in detail the construction of the hydraulic cylinders on the pulley B. Fig. 208 shows typical indicator diagrams from this

FIG. 205. — Flather's Hydraulic Transmission Dynamometer.

apparatus. Both were taken from a dynamometer connected to a mining drill. The first was taken when the drill was sharp, the second when it was dull.

Among the advantages claimed are: (1) its simplicity, (2) that it is not appreciably affected by the velocity of the shafting, (3) that no countershaft is required, (4) by connecting it to a recording gage a continuous diagram of the load can be obtained.

Torsion or Shaft Dynamometers. When a shaft is subjected to a twisting moment an angular twist is produced which is proportional to

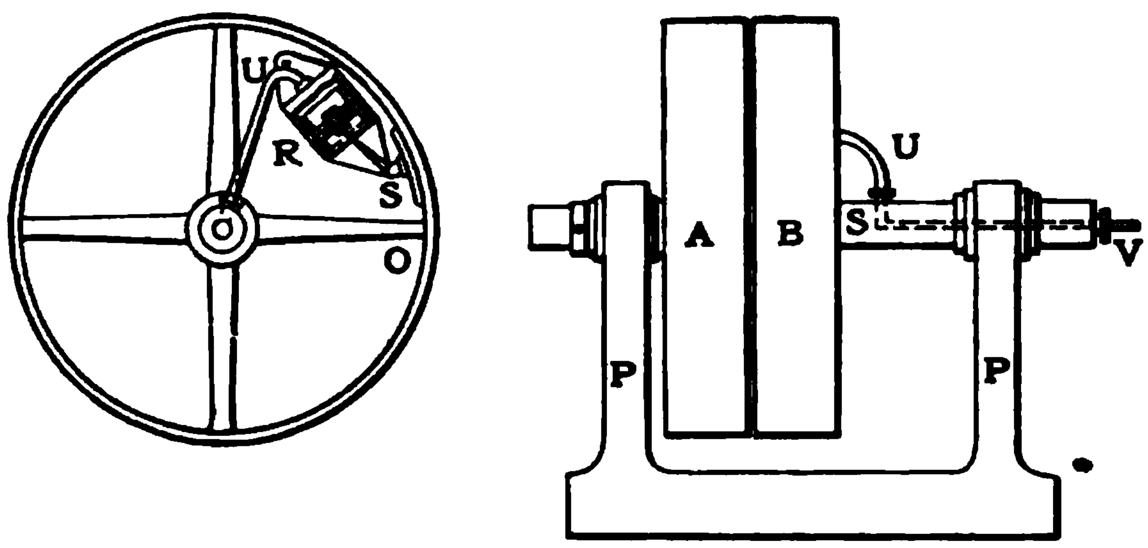


FIG. 206. — Diagram Showing Pulleys, Pistons, and Shaft of Flather's Dynamometer.

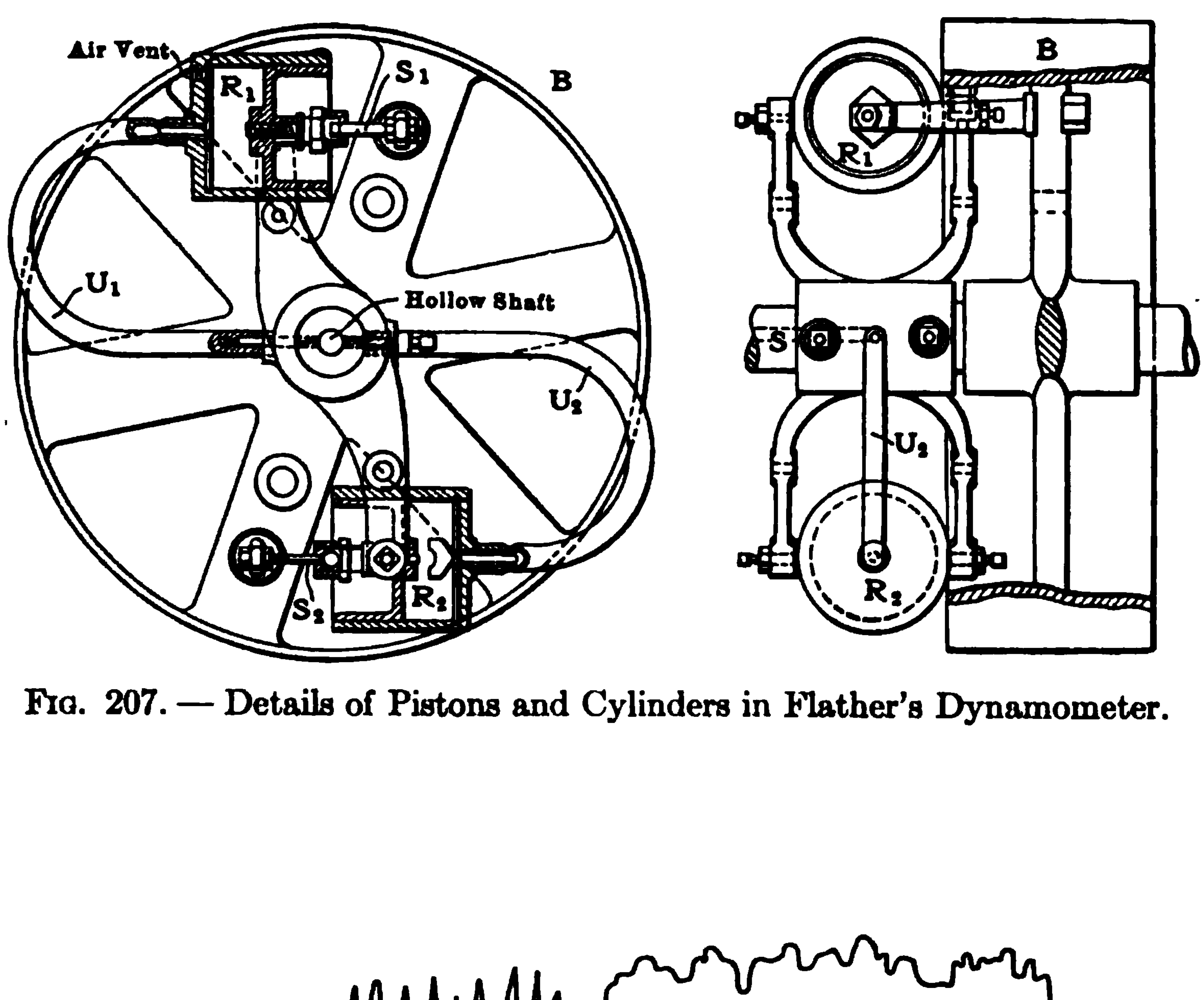


FIG. 207. — Details of Pistons and Cylinders in Flather's Dynamometer.

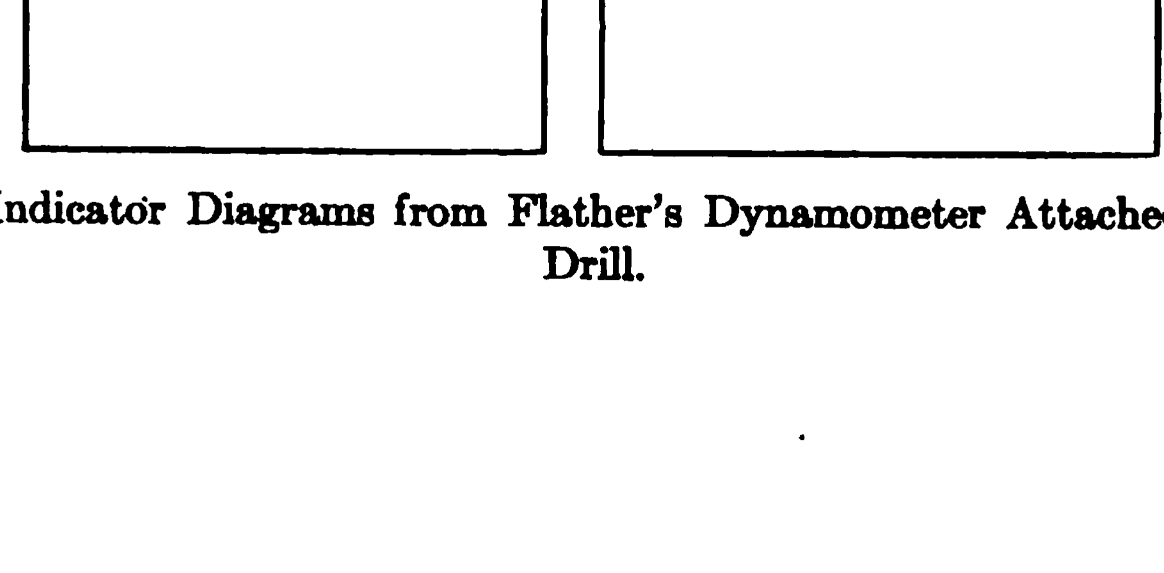


FIG. 208. — Indicator Diagrams from Flather's Dynamometer Attached to a Mining Drill.

that moment. Thus if θ is the angle of twist produced by a twisting moment T in inch-pounds and if n is r.p.m., then

$$\text{b.h.p.} = \frac{2\pi Tn}{12 \times 33{,}000}.$$

Torsion meters, shown in **Figs. 209, 210, 211** and **212**, although applicable to large as well as small powers, have their most important applications for measuring shaft horse power of marine turbines and engines.

A Shaft Dynamometer consists essentially of a long metal tube encircling the shaft and fastened to it at one end, but free at the other and is maintained in alignment by adjustable rollers; two radial arms, one attached to the shaft and the other to the free end of the tube, which rotate a slight amount with reference to each other, according to the twist of the enclosed length of shaft; and a set of levers which multiply this rotative movement and at the same time convert it into linear motion, which is transmitted to a sleeve and collar mounted upon the shaft and sliding thereon. These parts all revolve with the shaft. An independent indicating apparatus is provided, which is mounted on a stationary frame, and the sliding movements of the rotating collar is transmitted through it to an index hand. The torsional strain is determined from the reading of the accompanying scale, which is graduated to millimeters.

The zero reading is found by disconnecting the propeller and turning the shaft at a slow speed, first in one direction and then in the other, observing the indication in both cases, and fixing the point of zero strain at the mean of the two. When it is impracticable to disconnect the propeller the readings may be taken when the vessel is drifting under her own headway after shutting off steam. The calibration of the instrument, which can best be done when the shaft is in the shop before installation, is carried on by securing the shaft in a fixed position, and applying a torsional strain by means of weights at the end of a lever attached beyond the dynamometer, taking readings with a number of different weights.

The horse power shown by the dynamometer is determined by multiplying the reading of the instrument expressed in millimeters by the number of revolutions of the shaft per minute, and by a constant determined from the calibration. The constant is an expression for the horse power corresponding to a speed of one revolution per minute and a reading of one millimeter.

A shaft dynamometer requires very delicate adjustment as such instruments used on large steam turbines and engines requiring a shaft of comparatively large size, require a movement at the end of the two arms of only one hundredth of an inch to produce a change of 500 horse power in the load being transmitted.

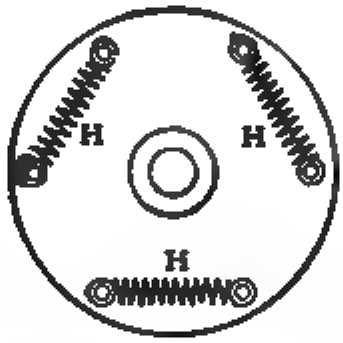


FIG. 209. — Spring Dynamometer.

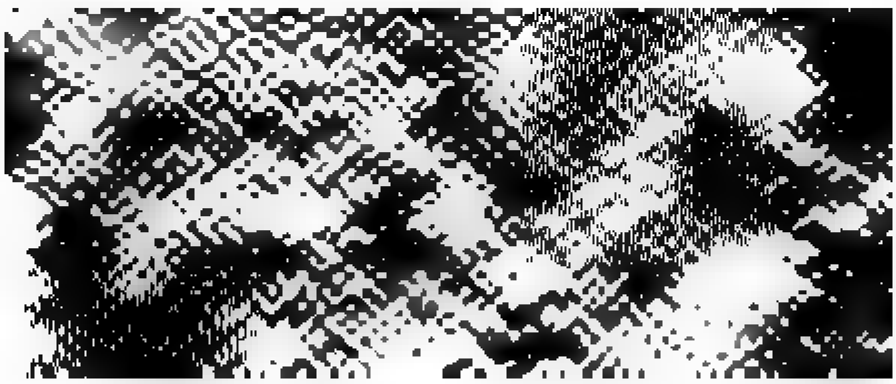


FIG. 210. — Mechanically Operated Shaft Dynamometer.

FIG. 211. — Electrically Operated Shaft Dynamometer.

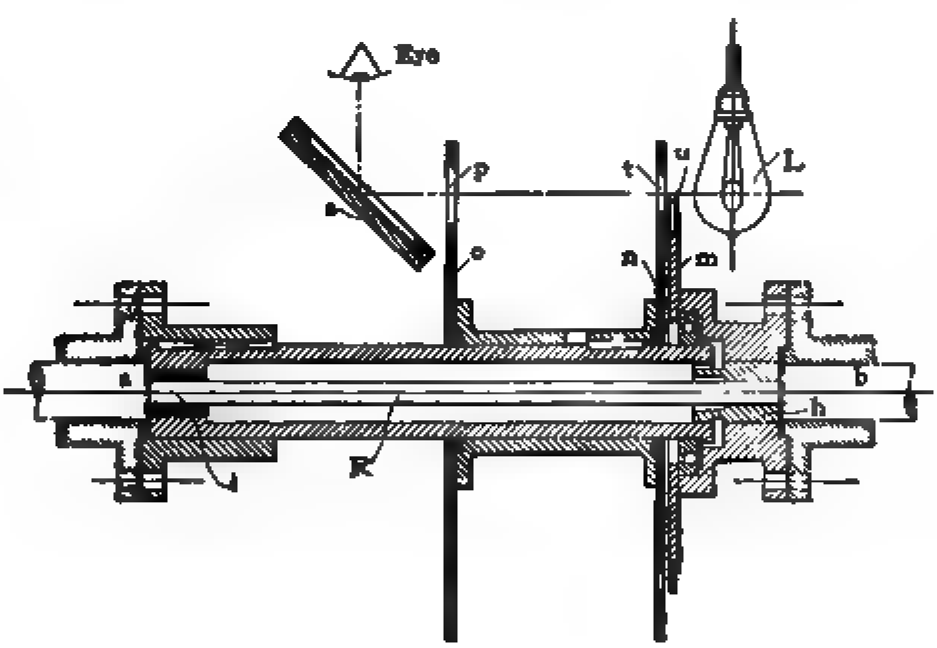


FIG. 212. — Shaft Dynamometer with Optical Means of Observation.

Kenerson Torsion Dynamometer[1] (**Figs. 213** and **214**) consists essentially of a divided shaft with a flanged coupling rigidly fastened to each of the adjoining ends. These flanges are only loosely connected by stud bolts and "latches." The latter are twisted by the power impressed so that their ends are forced against a pressure plate. The pressure against this plate is a measure of the power transmitted. Ball-bearing races attached to a diaphragm covering an oil chamber communicate this pressure or thrust to a chamber in which the pressure is indicated by a Bourdon gage. Readings must be corrected for static head if the gage is placed above or below the couplings.

Fig. 213. — Kenerson Dynamometer.

The **accelerometer** shown in **Fig. 215** is used a great deal in automobile testing. From the indications of this instrument in terms of acceleration the power developed can be computed.[2] This instrument is designed on the principle that a pendulum hung on a moving body will be in a vertical position when at rest and moving uniformly. In the figure **D** is a copper disk pivoted on a rotating vertical axis, **M** is a magnet for dampening the movement of this disk, and **G** shows two gear wheels of equal diameter, one fastened to the axis supporting the disk and the other on a separate axis carrying the needle **N**. A coil spring brings the needle back to zero. One side of the disk **D** is heavier than the other, so that when the acceleration takes place in

Fig. 214. — Parts of Kenerson Dynamometer.

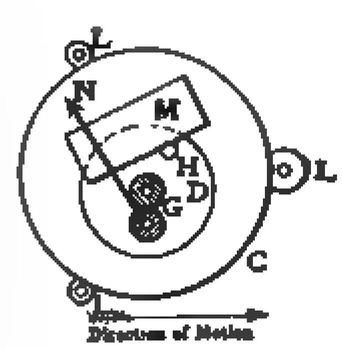


Fig. 215. — Accelerometer.

[1] *Transactions American Society of Mechanical Engineers*, vol. 31 (1909), pages 171 to 179.

[2] *Internal Combustion Engineering*, (London), Oct. 2, 1912, and *Engineering*, Sept. 16, 1910.

the direction of the arrow the heavier side tends to lag behind, causing a movement of the gears and the needle.

If **F** is the total resistance in pounds per ton at **N** miles per hour and **W** is the weight of the vehicle in tons, then brake horse power equals **F** × **W** × **N** divided by a constant.

CHAPTER VII

FLOW OF FLUIDS

THE flow of fluids will be discussed under these heads:

1. The flow of air.
2. The flow of steam.
3. The flow of water.

The Flow of Air. When subjected to only a low pressure, air and many other gases are usually measured by a **gas meter,** of which there are many types sold commercially. There are, however, two general types: (1) "**wet**" and (2) "**dry.**" The former is by far the more reliable and should be always used in preference to a "dry" meter when it can be obtained. "Wet" meters receive their name from the water seal maintained in them. This seal must be always kept at a constant level, determined by calibration, and before using such a meter in a test one should always observe whether the water level is at the standard mark. If it is not, then water must be added or withdrawn as the case may be. A section of a "wet" meter is shown in **Fig. 220.** It consists of a rotor somewhat resembling a paddle wheel revolving on a horizontal axis in an enclosed cylindrical casing partly filled with water. **Fig. 221** illustrates a typical apparatus of this kind, with four compartments **A, B, C** and **D.** When air or any gas flows[1] into one of the chambers of the meter it accumulates over the surface of the water and by its pressure raises the chamber until it is filled.

During rotation, by means of the water seal the central ports **a, b, c** and **d** are opened and closed. When these are open gas from the pipe at the gas inlet is admitted to the compartments out of the water. Discharge ports **a′, b′, c′** and **d′** through which the gas passes to the discharge pipe are also opened and closed by the water seal. If the drum revolves freely no gas can pass through the meter without producing the required movement of the recording mechanism, because the admission and discharge ports will not be open simultaneously in any compartment. Even the smallest rates of flow are accurately measured. Rotation is due to the difference in water level, as shown in the figure, between the two sides when the meter is operating. In the figure gas is being admitted on the left-hand side so that the pressure on that side will be slightly greater

[1] The nature or specific gravity of the gas is not important, as gas meters are calibrated to record volumes, usually cubic feet.

than on the right, causing the water level to be higher and making a greater weight on the right-hand side than on the left. On account of the

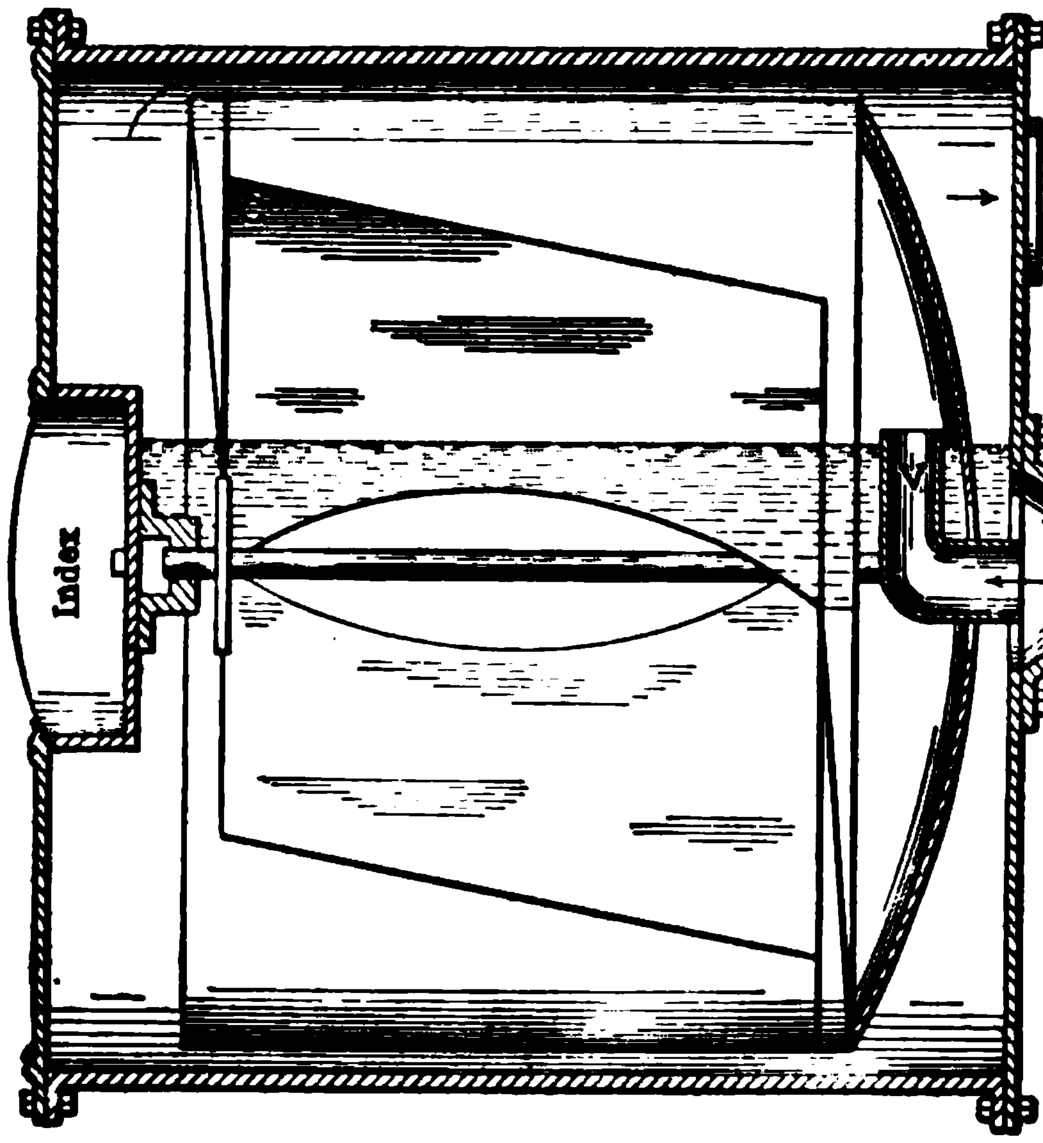


FIG. 220. — Typical "Wet" Gas Meter.

difficulty in making the admission and discharge ports of the rotor of sufficient size the admission ports are usually placed at one end and the dis-

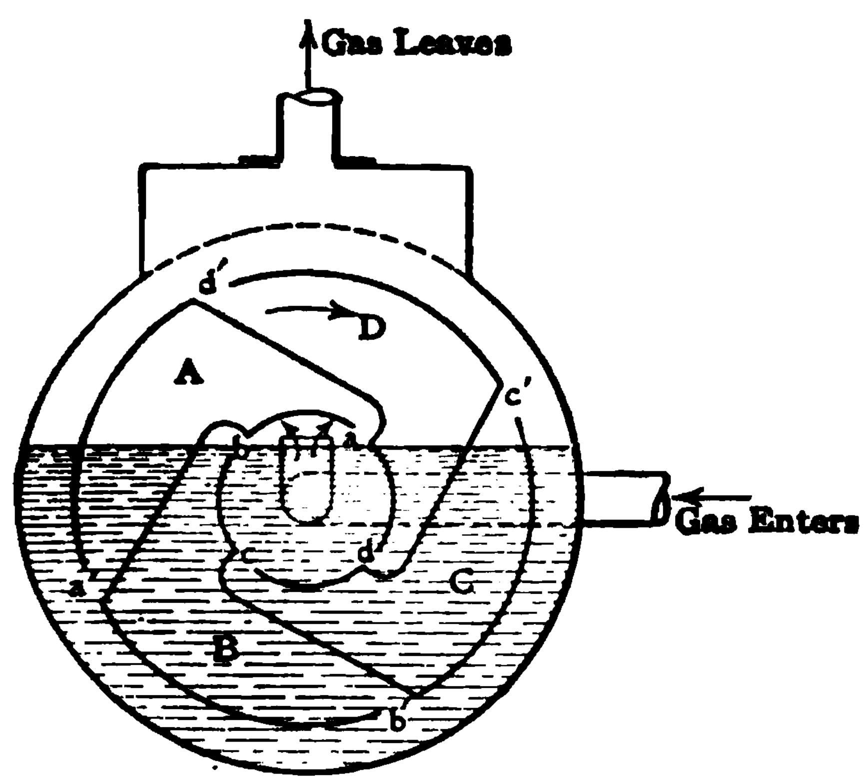


FIG. 221. — Diagram of "Wet" Gas Meter.

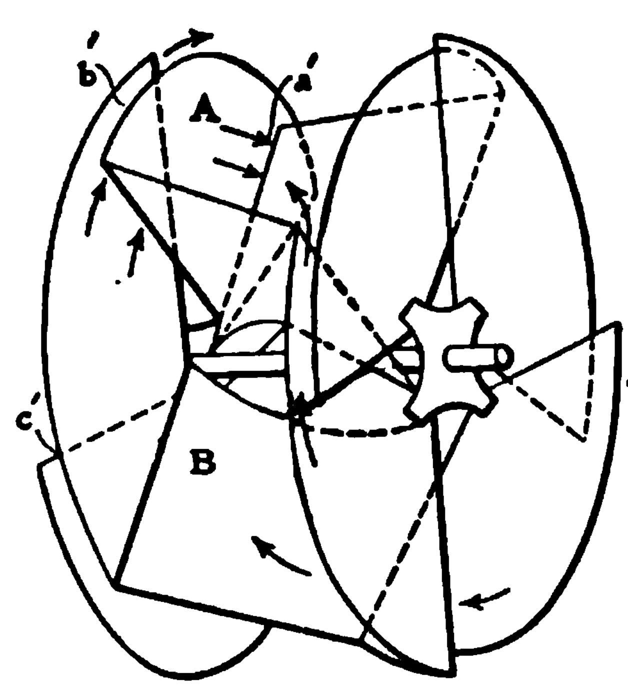


FIG. 222. — Rotor of "Wet" Gas Meter.

charge ports at the other, as illustrated in **Fig. 222**, the flow of gas being shown by the arrows. In the assembled view (**Fig. 220**) the gas enters at the dry-well, **V**, passes through the drum and out at the front end, then

over the drum between it and the case to the outlet.[1] In this way the drum is made to revolve to the left by the pressure on the surface of the water below and the slanted partition **C** above, forming an ever-increasing pyramidal space between the surface of the water and the plane of the slanted partition.

Fluctuations of pressure or of velocity cause errors only when great enough to produce a sufficient surging of the water, so that the water sealing on the valves may at times be prevented. If the flow is intermittent as in the suction pipe of gas engines and compressors a pressure regulator must be provided. A rubber bag is used as a regulator when the gas is under pressure and a diving bell hung on springs for suction gas.

" Dry " Gas Meters are used for the usual " house metering " of gas. They are not nearly so accurate as the " wet " types, but can be used more conveniently because they are not dependent on a constantly maintained water level. In simplest terms such meters consist of two chambers separated by a vertical partition, each chamber containing an interior measuring receiver having a flexible shell. Gas is admitted to these measuring receivers alternately by means of slide valves actuated automatically. The reciprocating movement of alternately filling and emptying these receivers operates the counting mechanism.

" Wet " meters are usually very accurate, while " dry " meters are not " supposed to be instruments of great accuracy."

Pitot Tube for Measurement of Air. Probably the most accurate method of measuring air in large volumes is by means of a Pitot tube. A standard instrument of this kind designed for the measurement of air by the American Blower Company is shown in **Fig. 223.**

It consists simply of two concentric brass tubes, a small one **A** being placed inside of a larger one **B**, as illustrated in detail in **Fig. 224.** These tubes are arranged so that each has a separate connection, as at **A′** and **B′**. The lower end of the small tube is open at **A**, while the outside and larger tube is tapered and closed; but approximately midway along its horizontal portion, as shown in the figure, there are two holes on each side.[2] These holes should be not much more than $\frac{1}{50}$ inch in diameter.

The Taylor Pitot tube was formerly much used. It differs essentially from the one shown in the figures by having a short slot $2\frac{1}{2}$ inches long and $\frac{1}{16}$ inch wide on each side. At high velocities these slots cause

[1] More frequently the outlet for the gas is on the top of the casing than at the back as shown in the figure.

[2] The holes for static pressure are shown here at the top and bottom. In practice these holes are usually placed at the sides of the tube as in this location they are less likely to become filled with dust and refuse.

eddies[1] and the static pressure observed will be too high. Taylor tubes being 13 inches long compared with $4\frac{1}{2}$ inches for the **ABC** types are much more clumsy and liable to breakage in handling.

When the instrument is used it is placed so that the opening at **A** points against the direction of flow and receives the full effect of the pressure due to the velocity of flow. The side openings in **B** are subjected to only

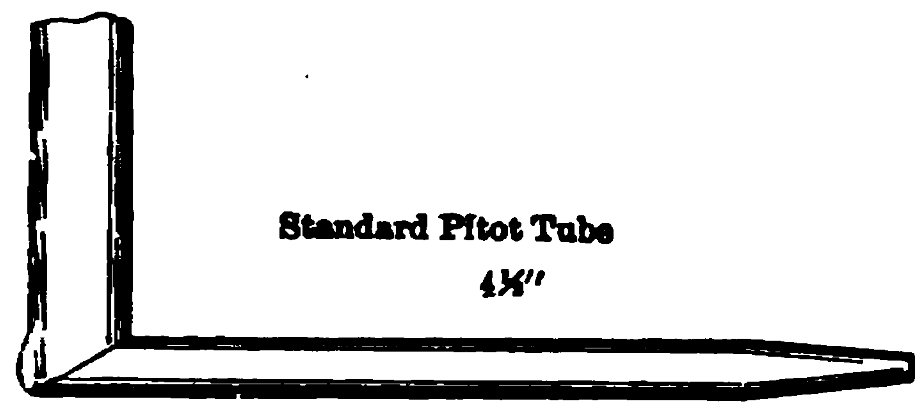


FIG. 223. — "American Blower Co." Standard Pitot Tube.

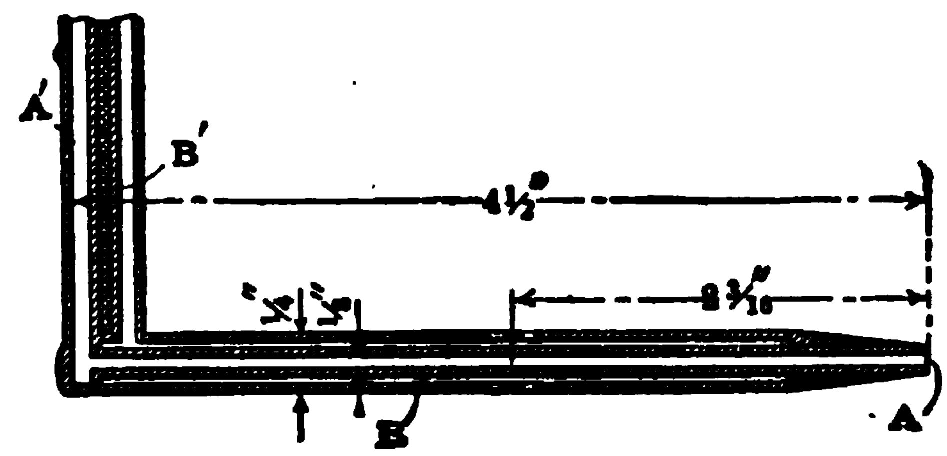


FIG. 224. — Section of Pitot Tube.

the static pressure. For convenience let **p** = velocity pressure and **s** = static pressure. For example, the difference in the levels in the manometer, **a**, **Fig. 226**, is therefore that due to (**p** + **s**) − **s**, or simply **p**, the **velocity pressure.**

Another type of Pitot tube much used and known as **Burnham's** is shown in **Fig. 225**. It is made up of two tubes **A** and **B**. Tube **A** is for obtaining the total pressure and in principle is not different from those already described. The static tube **B** is unique. It is open at the end and is pointed downward. On the side toward the direction of flow it is beveled at an angle of about 45 degrees. Neither the Taylor nor the Burnham types are satisfactory for measuring velocities above 6,000 to 8,000 feet per minute, while the **ABC** type is accurate for very high velocities. The latter type is the one recommended by the Power Test Committee of the A.S.M.E. (see *Journal*, Nov., 1912, page 1831).

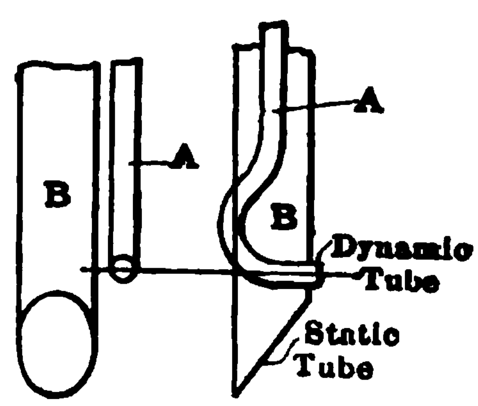


FIG. 225. — Burnham's Pitot Tube.

Pitot tubes are usually connected to manometers or preferably to sensitive draft gages, showing the pressure in small fractions of an inch of water. When the end of the Pitot tube at **A′** is connected to the left-hand end of a draft gage, like those in the figures on page 182, and the end at **B′** is attached to the right-hand end, the instrument acts as a differential gage and the difference between the reading when thus connected and its zero reading is the pressure in inches of water corre-

[1] This inaccuracy of the Taylor type of tube can be remedied by soldering neatly a sheet of fine brass gauze over the slots. For results of tests with the Taylor tube see *Trans. A.S.M.E.*, vol. 33 (1911), pages 1137–1173.

sponding to the velocity alone; that is $(p + s) - s$. If, as before, we call the velocity pressure **F** in inches of water, and if **h** is the height or " head " in **feet** of an equivalent column of air producing the same pressure, then the velocity of the air **v** in feet per **minute** is

$$v = 60\sqrt{2gh},$$

where **g** is the force of gravity (32.2), and

$$h = \frac{p}{12} \times \frac{\text{wt. of a cu. ft. of water}^{1}}{\text{wt. of a cu. ft. of air}}$$

$$= \frac{62.3\,p}{12 \times \text{wt. cu. ft. air}} = \frac{5.196\,p}{\text{wt. cu. ft. air}},$$

$$v = 1097\sqrt{\frac{p}{\text{wt. cu. ft. air}}}. \qquad (34)$$

In the following table the weight is given of dry air and also the weight of air completely saturated with moisture (100 per cent humidity). The data given are at atmospheric pressure (14.7 pounds per square inch) and the temperature given is that indicated by the "dry" thermometer. By interpolating between these tables, the weight of air for any temperature and degree of saturation is easily obtained. Remembering also that the weight per cubic foot is directly proportional to the absolute pressure, the weight for any pressure is readily determined. Tables for determining the percentage of saturation by means of wet- and dry-bulb thermometers are given on page 368.

For many engineering calculations relating to tests of fans and blowers it is accurate enough to interpolate between columns (3) and (4) to allow for humidity. For work requiring greater accuracy use the curve sheets given on pages 1006 and 1007 in the *Transactions of A.S.M.E.*, vol. 33. Observe that the numbers on the curved lines on page 1006 should be marked per cent instead of degrees.

To determine the volume of air flowing through a circular duct, the average velocity is most accurately obtained by dividing the cross-section of the duct or pipe into 5 or 10 imaginary annular rings of **equal area**. Each of these rings is then again divided into two others of equal area. Observations for a given flow are then made by shifting the Pitot tube rapidly along a diameter and taking readings with the tip of the tube at each of these last points of subdivision, called " stations." Average velocities and flows are then readily obtained from these observations. This is sometimes known as the " ten point " method. For a pipe of radius **r**, divided into five annular rings, the " stations "

[1] The weight of a cubic foot of water at about "room" temperature (about 70 deg. Fahr.) is about 62.3 pounds.

PROPERTIES OF AIR.[1]

Temp. by Dry Bulb, Deg. F. (1)	Weight of Water Vapor per Pound Pure Air, Lbs. (2)	Specific Volume, Cu. Ft. per Lb. Dry Air. (3)	Specific Volume, Cu. Ft. per Lb. 100% Saturated Air. (4)	Temp. by Dry Bulb, Deg. F. (1)	Weight of Water Vapor per Pound Pure Air, Lbs. (2)	Specific Volume, Cu. Ft. per Lb. Dry Air. (3)	Specific Volume, Cu. Ft. per Lb. 100% Saturated Air. (4)
0	.0009	11.588	11.603	74	.0179	13.449	13.593
10	.0016	11.795	11.820	76	.0192	13.499	13.654
20	.0024	12.051	12.091	78	.0206	13.549	13.715
32	.0038	12.388	12.414	80	.0220	13.600	13.777
34	.0041	12.439	12.469	82	.0235	13.650	13.841
36	.0044	12.489	12.523	84	.0252	13.701	13.906
38	.0047	12.539	12.576	86	.0269	13.752	13.971
40	.0051	12.590	12.629	88	.0288	13.801	14.038
42	.0055	12.640	12.682	90	.0307	13.852	14.106
44	.0060	12.692	12.736	92	.0328	13.903	14.173
46	.0065	12.741	12.791	94	.0350	13.954	14.241
48	.0070	12.792	12.846	96	.0374	14.004	14.310
50	.0076	12.842	12.901	98	.0399	14.055	14.382
52	.0082	12.893	12.957	100	.0424	14.106	14.455
54	.0088	12.944	13.012	105	.0500	14.232	14.643
56	.0094	12.993	13.068	110	.0586	14.358	14.840
58	.0100	13.044	13.124	115	.0687	14.484	15.050
60	.0108	13.095	13.180	120	.0804	14.611	15.272
62	.0117	13.146	13.240	125	.0941	14.736	15.509
64	.0126	13.196	13.298	130	.1102	14.863	15.761
66	.0135	13.246	13.354	135	.1293	14.959	16.032
68	.0145	13.298	13.413	140	.1515	15.116	16.325
70	.0156	13.348	13.471	145	.1782	15.242	16.643
72	.0167	13.398	13.532	150	.2100	15.368	16.993

[1] W. H. Carrier in the *Transactions of A.S.M.E.*, vol. 33 (1911) pages 1005–1136. These table are generally considered more reliable than any other available data. The tables of the U. S. Weather Bureau are in error because they were computed on the assumption that saturated air is a perfect gas. The fallacy is particularly observable at high temperatures.

or points of observation would be at the following distances from the center: (1) 0.316 r; (2) 0.548 r; (3) 0.707 r; (4) 0.837 r; (5) 0.949 r.

When the Burnham Pitot tube is used readings are taken usually in only one position. This position giving the average velocity is stated to be at a distance of $\frac{8}{10}$ of the actual internal radius from the center of the pipe.

Ducts of square or rectangular section are usually divided up similarly into a series of elementary squares or rectangles.

Figs. 226 and **227** show the methods of connecting a Pitot tube to manometers for observing velocities when the pressure is above or below atmospheric. The usual case is where the pressure is greater than atmospheric, and the cases where it is less are most often in the suction line of a ventilating fan. In **Fig. 227** positions of "stations" (**a, b, c, and d**) are

marked on a board to assist in taking observations. Measurements of velocity with a Pitot tube should not be attempted if there is not at least 15 feet of straight pipe in the direction in which the tube is pointed. This precaution is necessary to avoid the effect of eddies in the pipe.

For Pressures above Atmospheric

FIG. 226.

For Pressures less than Atmospheric

FIG. 227.

Arrangement of Connections for Pitot Tube Measurements.

Anemometers. A very convenient and simple method for measuring directly the volume of the air, or any gases, is by using an instrument called an anemometer. This instrument, **Fig. 228,** consists in its essential parts of a light vane wheel like a screw-propeller having either flat or lightly curved vanes mounted on slender arms. The wheel must be made very light in weight, must be accurately balanced, and should move easily in its bearings. By its own motion it operates a counting mechanism attached to its shaft which indicates velocities in feet. Readings of the counter are taken at the beginning and end of a suitable lapse of time, usually ½ to 1 minute. Such instruments must be placed with the axis of rotation in the direction of the flow of air or gas. They have upper and lower velocity limits beyond which they should not be used. The lower limit cannot be defined as it will depend on the precautions taken in manufacture and in use to eliminate friction. As regards the higher limit, it will usually depend on the size of the wheel, large wheels being less suitable than smaller ones for high velocities. Practically none should be used for velocities higher than 1,000 feet per minute.

FIG. 228. — A Typical Anemometer for Measuring Velocity of Air.

A calibration chart must always be provided for such instruments and the calibration should be frequently checked at several points within its velocity limits.

Methods of Calibrating Anemometers, Pitot Tubes, and Gas Meters. Probably the best method of calibrating anemometers is to compare their readings with the actual measurements of air discharged from a gasometer or gas-holder like Fig. 229. It consists of a tank A for holding water or other liquid into which the "bell" B is raised and lowered. The piping as shown is arranged with a three-way cock (see page 139), but for accurate work separate inlet and discharge pipes should be provided. The weight W is to counterbalance the weight of the bell. This method of counterbalancing if used without a means for correction would cause a change of pressure of the gas in the holder as the bell ascends or descends, due to a variation of depth of immersion. As a means to correct this, a compensating weight w is suspended from a cord wrapping over a cam C. Observations required to determine volume of gas are (1) pressure (usually read with a water manometer M) (2) temperature; and (3) movement of the bell in a given time. In order to get the temperature of the gas in the bell accurately the temperature of the liquid should be as nearly as possible the same as that of the gas. Movement of the bell is usually read directly with the help of a suitable scale and some form of sighting device to insure accuracy in reading the scale.

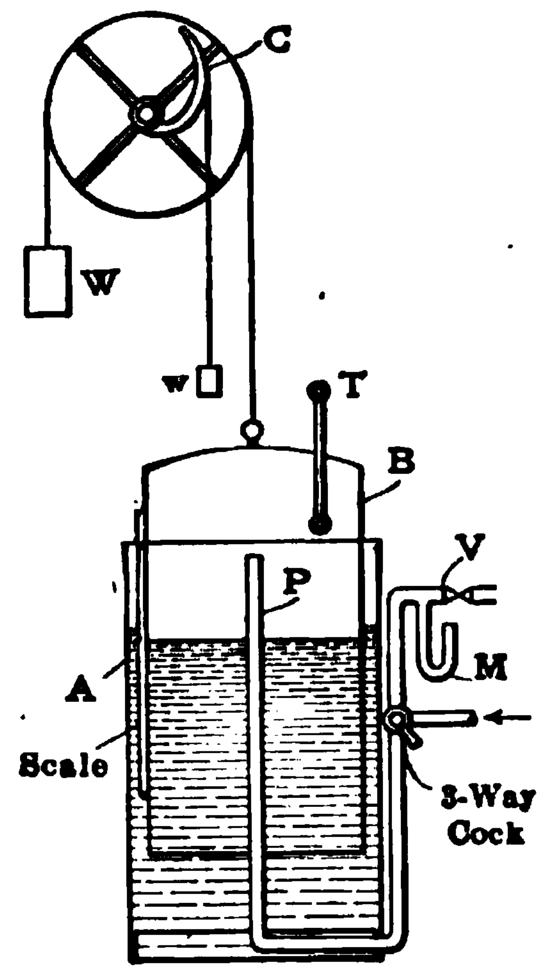


FIG. 229. — Gasometer.

Gasometers may be calibrated by calculation or by actual tests, preferably on the displacement of water. When the method of calculation is used, measurements of the diameter should be made at several points on the circumference to allow for possible lack of symmetry.

Large gasometers such as are installed at gas works are also frequently used to calibrate Pitot tubes; and gas meters are invariably calibrated by comparison with a gasometer.

Gas meters may also be calibrated by any apparatus suitable for the displacement of the gas as it is withdrawn by water or other suitable liquid. It is very necessary, of course, that when the weighings are made the pressure and temperature of the gas be accurately determined.

Anemometers are suitable only for low velocities (from about 50 to 1500 feet per minute) and Pitot tubes in general are best adapted to

velocities from 300 to 3000 feet per minute, but instruments like the "A. B. C." type (**Fig. 223**) designed to avoid eddies around the "static" openings are satisfactory up to 6000 feet per minute.

In order to use an anemometer successfully all the gas to be measured must be passed through openings of suitable size in which the instrument can be placed. These openings should each have an area of about 15 square inches (if the anemometer is about 2½ inches in diameter) so that the resistance interposed by the instrument will be negligible.

A very common method of calibrating anemometers and Pitot tubes is by mounting them on the end of a long and light rod arranged to be revolved about a central point. The readings of the instrument are compared with its computed velocity. Other methods of calibration under more nearly "working conditions" are generally considered better, as the method of swinging about a central point makes the instrument read too high on account of the eddies produced.

"The standard of reference for calibrating Pitot tubes, anemometers, etc., is the gasometer,[1] and if the instrument used cannot be calibrated by actual measurement, the constants employed should be those obtained from a similar instrument which has been calibrated by actual reference to a gasometer measurement." (Report of Power Test Committee, A.S.M.E., Nov., 1912.)

Flow of Air through an Orifice. Air under comparatively high pressures is usually measured in practice by means of pressure and tempera-

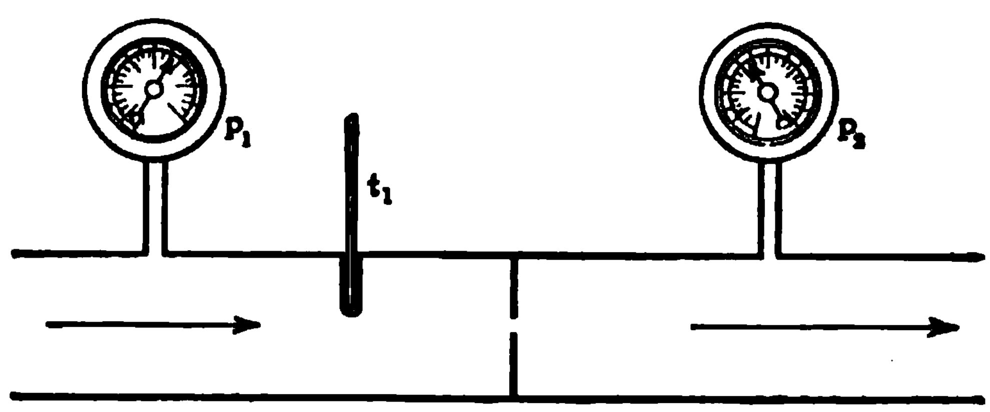


Fig. 230. — Measuring Flow of Air through an Orifice.

ture observations made on the two sides of a sharp-edged orifice in a diaphragm. **Fig. 230** illustrates the method with two pressure gages on opposite sides of the orifice and a thermometer for obtaining the tem-

[1] It is very difficult to get a uniform temperature in a gasometer so that the volume of gas discharging from the outlet may be different by 2 or 3 per cent from that indicated by the scale on the gasometer. When accurate work is to be done a better method to use is the displacement of water as measured by its weight in a vessel of heavy sheet metal. The walls of the vessel should be heavy to prevent rapid radiation to the surrounding air. A long thermometer should be used which should be inside the vessel itself and may be read through a peep-hole. If the vessel is of heavy glass the conditions will be still better.

perature t_1 at the initial or higher pressure p_1. The flow of air **w**, in pounds per second, may then be calculated by **Fliegner's formulas.**

$$w = .530 \times a \frac{p_1}{\sqrt{T_1}} \text{ when } p_1 \text{ is greater than } 2\,p_2. \quad . \quad . \quad . \quad . \quad (45)$$

$$w = 1.060 \times a \sqrt{\frac{p_2\,(p_1 - p_2)}{T_1}} \text{ when } p_1 \text{ is less than } 2\,p_2, \quad . \quad . \quad (46)$$

where **a** is the area of the orifice in **square inches,** T_1 is the absolute initial temperature in degrees Fahrenheit at the absolute pressure p_1 in the "reservoir or high-pressure side" and p_2 is the absolute discharge pressure, both in pounds per **square inch.** When the discharge from the orifice is directly into the atmosphere, p_2 is obviously barometric pressure.

For small pressures it is often desirable to substitute manometers for pressure gages. One leg of a U-tube manometer can be connected to the high-pressure side of the orifice and the other leg to the low-pressure side. Many engineers insert valves or cocks between the manometer and the pipe in which the pressure is to be observed for the purpose of "dampening" oscillations. This practice is not to be recommended as there is always the possibility that the pressure is being throttled.[1] A better method is to use a U-tube made with a restricted area at the bend between the two legs. This will reduce oscillations and not affect the accuracy of the observations.

Discharge from compressors and the air supply for gas engines are frequently measured by orifice methods.

When $p_1 - p_2$ is small compared with p_1, the simple law of discharge[2] of fluids can be used as follows:

$$w = \frac{fa}{144} \sqrt{2\,g \times 144\,(p_1 - p_2)\,s}, \quad . \quad . \quad . \quad . \quad . \quad (47)$$

[1] Report of Power Test Committee, *Journal A.S.M.E.*, Nov., 1912, page 1695.

[2] If the density is fairly constant,

$$\frac{144\,p_1}{s} + \frac{v_1^2}{2\,g} = \frac{144\,p_2}{s} + \frac{v_0^2}{2\,g},$$

where v_1 is the velocity in feet per second in the "approach" to the orifice and v_0 is the velocity in the orifice itself. Since v_1 should be very small compared with v_0,

$$\frac{v_0^2}{2\,g} = \frac{144\,(p_1 - p_2)}{s},$$

$$v_0 = \sqrt{\frac{2\,g \times 144\,(p_1 - p_2)}{s}}.$$

$$w = \frac{fav_0s}{144} = fas\sqrt{\frac{2\,g \times 144\,(p_1 - p_2)}{s}},$$

or

$$w = \frac{fa}{144}\sqrt{2\,g \times 144\,(p_1 - p_2)\,s}.$$

where w is the weight in pounds discharged per second, a is the area of smallest section of orifice in square inches, the pressures p_1 and p_2 are in pounds per square inch, f is a coefficient from experiments, g is the acceleration due to gravity (32.2), and s is the unit weight of the gas measured, in pounds per cubic foot, for the average of the initial and final conditions of temperature and pressure (Table of weight of air on page 181). If the difference in pressure is measured in inches of water **h** with a manometer then

$$144 (p_1 - p_2) = \frac{62.4}{12} \times h, \quad \text{(lbs. per sq. ft.)}$$

$$w = \frac{fa}{144}\sqrt{2\,ghs \times \frac{62.4}{12}}, \quad \text{(lbs. per sec.),}$$

where 62.4 is the weight of a cubic feet of water (density) at usual "room" temperatures.

This equation can also be transformed so that a table of the weight of air is not needed, since by elementary thermodynamics $144\,pv = 53.3\,T$, where v is the volume in cubic feet of one pound of air or other gas and T is the absolute temperature in degrees Fahrenheit. Since v is the reciprocal of s, then

$$s = 144\,p \div 53.3\,T, \text{ and}$$

$$w = .209\,fa\sqrt{\frac{hp}{T_1}}. \qquad (48)$$

Here p and T should be the values obtained by averaging the initial and final pressures and temperatures. Great care should be exercised in obtaining correct temperatures. When equation (47) is used, for accurate work, corrections of s for humidity must be made.[1]

For measurements made with orifices with a well-rounded entrance and a smooth bore so that there is practically no contraction of the jet the coefficient f in equations (47) and (48) may be taken as 0.98. In the rounding portion of the entrance to such a nozzle the largest diameter must be at least twice the diameter of the smallest section. For circular orifices with sharp corners Professor Dalby[2] in reporting very recent experiments stated that the coefficient for his sharp-edged orifices in a thin plate of various sizes from 1 inch to 5 inches in diameter was in all cases approximately .60; and these data agree very well with those published by Durley.[3]

[1] Tables of humidity are given on page 368.

[2] *Engineering* (London), Sept. 9, 1910, page 380, and Ashcroft in *Proc. Institution of Civil Engineers*, vol. 173, page 289.

[3] *Transactions American Society of Mechanical Engineers*, vol. 27 (1905), page 193.

When $p_2 \div p_1 = .99$ the values obtained with this coefficient are in error less than $\frac{1}{2}$ per cent; and when $p_2 \div p_1 = .93$ the error is less than 2 per cent.

Flow of Air Measured by Cooling. This method depends on taking from the air an amount of heat[1] which can be measured and then computing from the heat units absorbed, the difference in temperature, and specific heat of the air, its weight and volume.[2] The arrangement of the apparatus is shown in **Fig. 231.** A coil of pipes **C**, of which the cooling surface is as equally as possible distributed over the section of the duct **D, D′**, carrying the air to be measured, is used to absorb heat by circulating water through it. Thermometers are arranged so that the temperatures of the air and of the water can be observed, and a

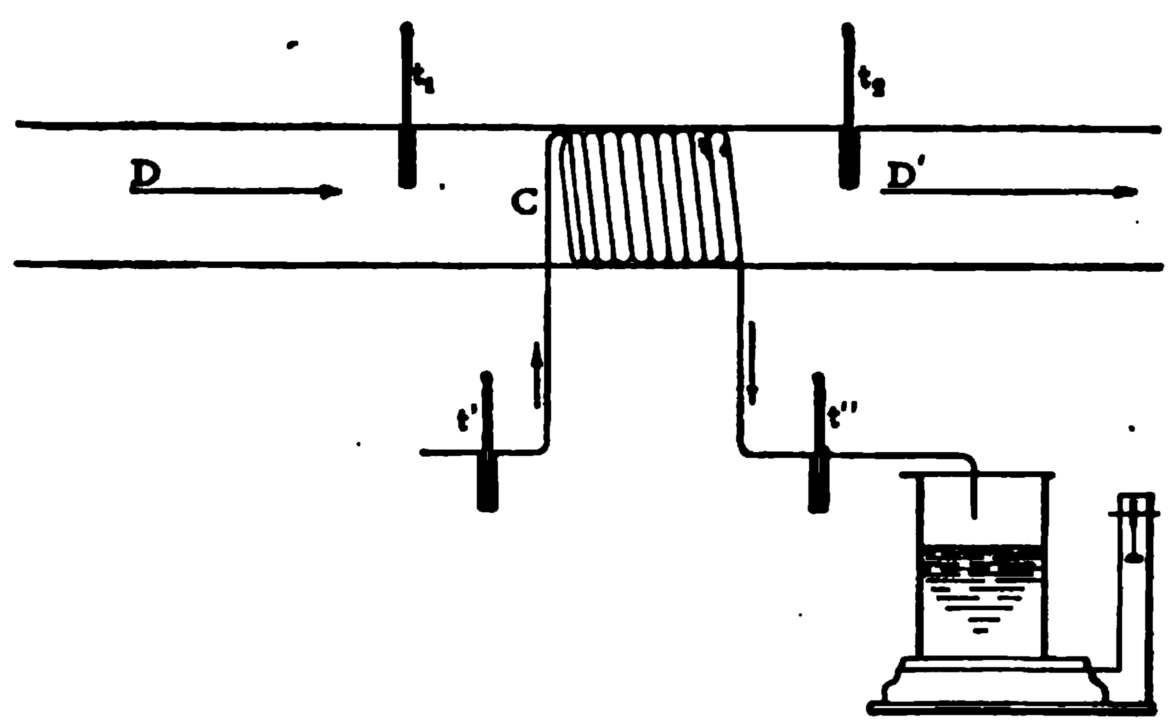


FIG. 231. — Measuring Flow of Air by Cooling.

platform scales is shown for obtaining the weight of water. Using the symbols t_1 and t_2 for the initial and final temperatures of the air in degrees Fahrenheit, t' and t'' for the temperatures of the water entering and leaving in degrees Fahrenheit, w_a = weight of air passing through duct in pounds per second, w_0 = weight of water collected in pounds per second, and .2375 = specific heat of the air at constant pressure and at temperatures not much above "atmospheric," then the heat absorbed by the water per second is $w_0 (t'' - t')$, B.t.u. and this equals the heat lost by the air, or $.2375\, w_a (t_1 - t_2)$, and therefore

$$w_a = \frac{w_0 (t'' - t')}{.2375 (t_1 - t_2)} = \frac{4.211\, w_0 (t'' - t')}{(t_1 - t_2)}. \quad . \quad . \quad . \quad . \quad (49)$$

[1] The method will be equally applicable if heat is added, as for example by passing steam through the coil. This method is often used to calibrate Pitot tubes and anemometers.

[2] The weight of a cubic foot of air is .0765 pound at 62 (522 abs.) degrees Fahrenheit and 14.7 pounds per square inch pressure. Since the volume is directly proportional to the absolute temperature, the weight at any other temperature is easily computed.

Thomas' electric meters employ this method of measuring gas, using electric current for the heating medium. The gas flows past a series of electrical resistances enclosed in a pipe. Electric heating by these resistances raises the temperature of the gas a few degrees. Temperatures before and after heating are measured by delicate resistance thermometers connected to a Wheatstone bridge and a sensitive galvanometer.

If the electrical energy used for heating be kept constant, and if the specific heat of the gas does not vary, the flow is inversely proportional to the rise in temperature.

If **E** is the amount of energy in watts, supplied to raise the temperature of **w** pounds of gas per hour through **t** degrees Fahrenheit and **C** is the specific heat of the gas at constant pressure, then

$$w = \frac{3.412\,E}{ts} \qquad (50)$$

If the volume of gas is required, its pressure, temperature and relative humidity must be determined.

Receiver Method of Measuring Air. None of the preceding methods are adaptable for measuring the volume of air at high pressures as in the case of measuring the discharge in tests of air compressors. Pumping air into a suitably strong receiver is a method often used. The compressor is made to pump against any desired pressure which is kept constant by a regulating valve on the discharge pipe:

P_1 and P_2 = absolute initial and final pressures for the test, pounds per square inch.

T_1 and T_2 = mean absolute initial and final temperatures, degrees Fahrenheit.

W_1 and W_2 = initial and final weight of air in the receiver, pounds.

V = volume of receiver, cubic feet.

$P_1V = WRT_1$, and $P_2V = W_2RT_2$, where **R** is the constant 53.3 for air, then weight of air pumped,

$$W_2 - W_1 = \frac{V}{53.3}\left(\frac{P_2}{T_2} - \frac{P_1}{T_1}\right). \qquad (51)$$

In accurate laboratory tests the humidity of the air entering the compressor should be measured in order to reduce this weight of air to the corresponding equivalent volume at atmospheric pressure and temperature.

Principal errors in this method are due to difficulty in measuring the average temperature in the receiver. Whenever practicable the final pressure should be maintained in the receiver at the end of the test until the final temperature is fairly constant.

The above method is often reversed by discharging air at high pressure from a receiver. Constant discharge pressure is maintained by throttling with a valve.

Measurement of Air by Chemical Analysis. Quantity of air supplied for combustion in boilers and other furnaces can be determined from the analysis of the products of combustion by the formulas given on pages 251 and 281.

Venturi Meters (see page 199) are also used successfully for measuring large volumes of gas as in tests of gas producers,[1] etc.

The Flow of Steam through Nozzles and Orifices. The flow of the steam from an orifice or nozzle has a very definite critical value when the final pressure is approximately 0.58 of the initial pressure. When the final pressure is less than this critical value the flow is expressed very accurately by the following empirical formula, based on the experiments of Professors Emswiler and Fessenden. Using the following symbols:

p_1 = initial absolute pressure of the steam in pounds per square inch;
p_2 = final absolute pressure of steam in pounds per square inch;
a = area of the smallest section of the nozzle or orifice in square inches.

Then the weight of the dry saturated steam discharged in pounds per second is approximately,[2]

$$w = \frac{p_1^{.97}a}{60.5} \text{ when } p_2 \text{ is less than } 0.58\, p_1. \quad . \quad . \quad . \quad . \quad (52)$$

Now since in the theoretical formulas the weight discharged is inversely proportional to the square root of the specific volume v, or w is proportional to $\sqrt{\frac{p_1}{v}}$ the formula above corrected for initial quality x of the steam is

$$w = \frac{p_1^{.97}a}{60.5\sqrt{x}} \text{ when } p_2 \text{ is less than } 0.58\, p_1. \quad . \quad . \quad (54)$$

When the steam is **superheated** the specific volume is considerably increased, and for this condition the author has found that the following equation gives very satisfactory results,[3]

$$w = \frac{p_1^{.97}a}{60.5\,(1 + .00065\,d)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

[1] *Trans. A.S.M.E.*, vol. 28, page 483 and vol. 29, page 952. See also Bulletin No. 76, Builder's Iron Foundry, Providence, R. I.

[2] A somewhat simpler formula, known as **Napier's formula**, which is accurate enough for most calculations, is the following:

$$w = \frac{p_1 a}{70} \text{ when } p_2 \text{ is less than } 0.58\, p_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad (53)$$

[3] For a more extended discussion of the flow of steam see *The Steam Turbine*, by the author, pages 52–57.

when, as before, p_2 is less than $0.58\,p_1$ and where **d** is the number of degrees (Fahrenheit) of superheat.

When the final pressure p_2 is greater than $0.58\,p_1$, the formulas must be modified to correspond to the reduced flow observed by inserting a coefficient **K** as a factor in the right-hand member of the equations. Values of this coefficient are most conveniently obtained from the curve in **Fig. 232**, which was plotted from the experimental results obtained by Professor Rateau.

Formulas (52) to (55) are for the flow through nozzles with smooth walls, being well rounded at the entrance and the length along the axis

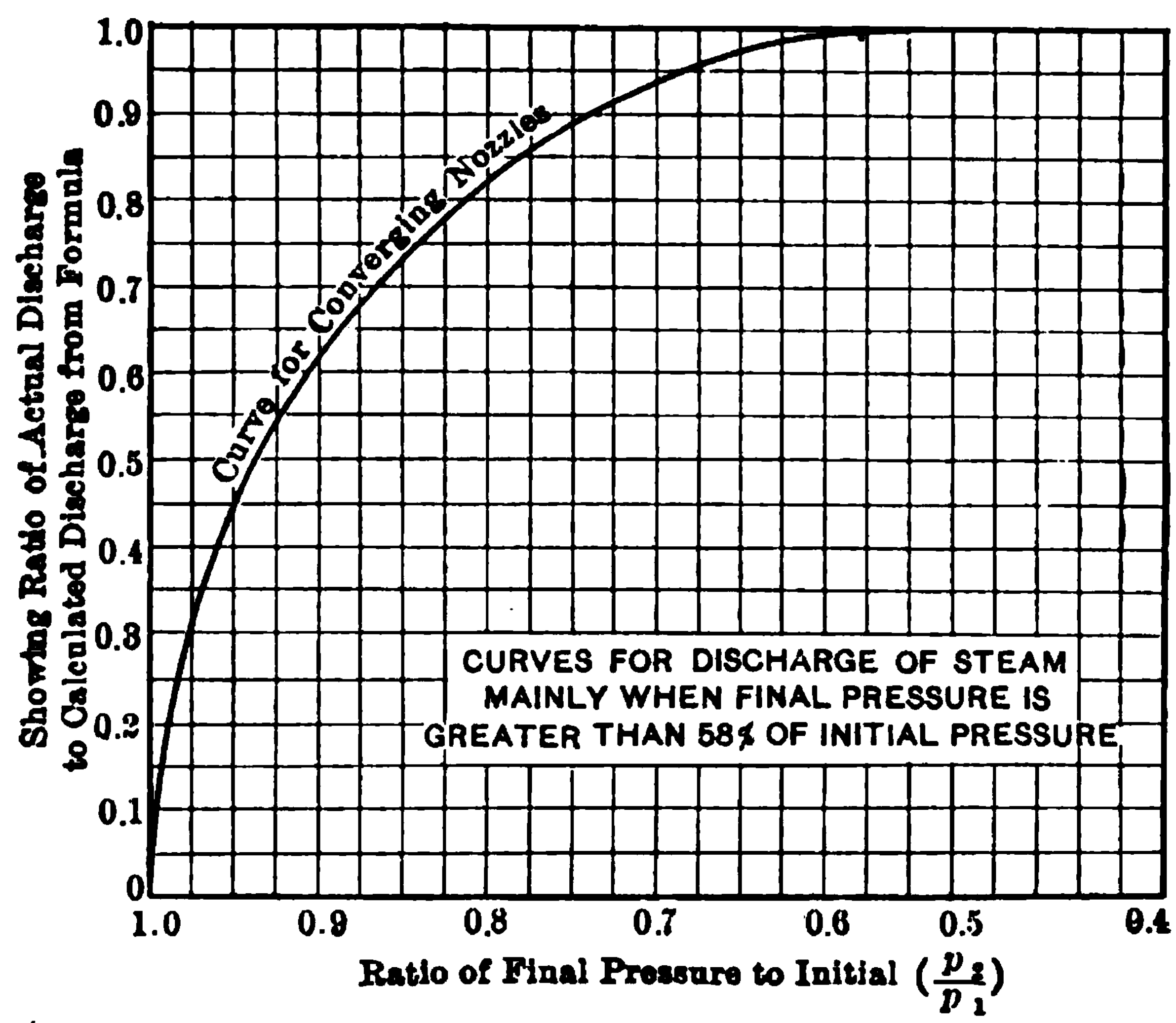


FIG. 232. — Rateau's Curve for Flow of Steam giving Values of the Coefficient K.

at least three times the length of the shortest side or diameter. If the nozzles are of approximately rectangular section they must be made without well-defined edges; in other words the cross-section must show well-rounded corners.

Orifice Measurements of the flow of steam are particularly recommended by some engineers for ascertaining the steam consumption of the "auxiliaries" in a power plant. This method commends itself particularly because of its simplicity and accuracy. It is best applied by inserting a plate $\frac{1}{8}$ inch thick with an orifice one inch in diameter, with **square** edges, at its center, between the two halves of a pair of flanges on the pipe through which the steam passes. Accurately calibrated steam gages are required on each side of the orifice to determine the loss of pressure. The weight of steam for the various differences of pressure may be determined by arranging the apparatus so that the

steam passing through the orifice will be discharged into a tank of water placed on a platform scales. The flow through this orifice in pounds of dry saturated steam per hour when the discharge pressure at the orifice is 100 pounds by the gage is given by the following table:[1]

Pressure, Drop, Lbs. per Sq. In.	Flow of Dry Steam per Hour, Lbs.	Pressure, Drop, Lbs. per Sq. In.	Flow of Dry Steam per Hour, Lbs.
½	430	5	1560
1	615	10	2180
2	930	15	2640
3	1200	20	3050
4	1400		

The Flow of Steam. Pitot Tube Meters are represented at their best in one of the types made by the General Electric Co. A nozzle plug (**Fig. 233**) is inserted into the steam pipe and in this plug there are two sets of holes each communicating with a separate tube starting from the end of the plug. These pipes are connected separately to the unions in **Fig. 234**, showing the apparatus. The "leading" set of holes is subjected to velocity plus static pressure, while the trailing holes are subjected to velocity less static pressure only. The principles of operation are therefore the same as for the measurement of air by the Pitot tube

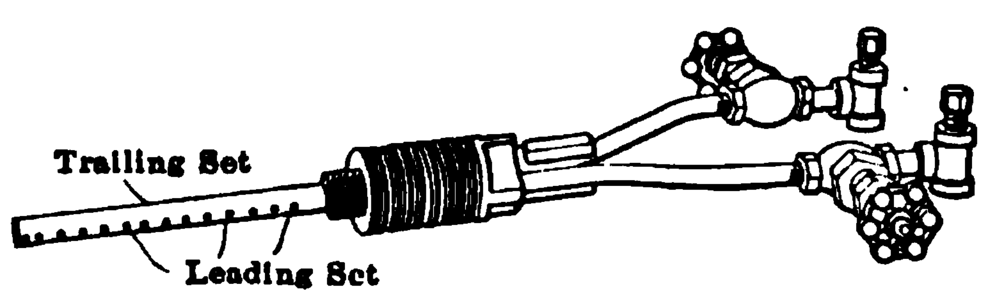


FIG. 233. — Nozzle Plug for Steam Meter.

(see page 178). The part actuating the recorder consists of two cups connected by a hollow tube forming an elongated U-shaped vessel which is filled with mercury. This vessel is balanced on knife-edges. The cups are connected by flexible steel tubing to the unions shown at the top of the figure which are to be joined to pipes running to the nozzle plug. In the operation of the instrument the excess of pressure in the leading holes of the nozzle plug causes the mercury to shift its level and stand higher in the cup connected to the trailing holes. As a result of this unbalancing of weights the whole vessel will swing on the knife edges untii equilibrium is again established. A clock and drum device is provided for recording on charts the movement of the mercury vessel on its knife edges. Charts are graduated in pounds of steam per hour. Correction for variation of flow on account of fluctua-

[1] *Journal A.S.M.E.*, Nov. 1912, page 1693.

tion of pressure is automatic. This correction device is simply a Bourdon tube of a pressure gage connected to the recording device so that as the curvature of the tube changes to correspond with the pressure it shifts a small weight intended to adjust the pen. When superheated

To Nozzle Plug

Fig. 234. —Steam Meter of Pitot Tube Type.

steam is being measured temperature correction must be made by shifting the same weight by hand.

The Burnham Steam Meter is one of the simplest types, and is serviceable only as an indicating instrument for "rough" measurements. The difference in level between the tip of the Pitot tube and the water in a gage glass is proportional to the flow of steam. The Pitot Tube used is shown in **Fig. 225**.

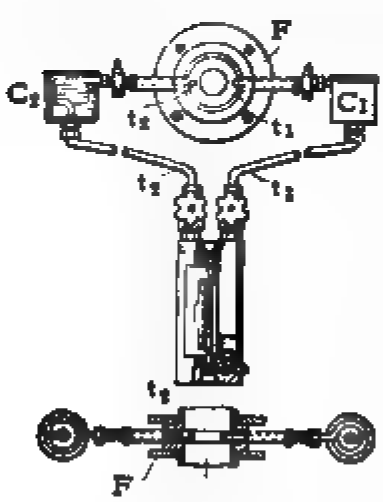


Fig. 236. — Orifice Steam Meter.

The **Orifice Steam Meter (Fig. 236)** requires the insertion into the steam pipe of a special flanged fitting **F** in which there is an orifice as shown. Pressure difference between the two tubes t_1 and t_2 located in this flange (produced by velocity) is measured by a differential mercury manometer. Without changing the orifice the apparatus is not adapted to a large range in the rate of flow. The spiral coils c_1 and c_2 are inserted for maintaining by condensation constant water

levels in each of the legs of the manometer, irrespective of variations of pressure.

A variation of the orifice method has been applied very successfully on steam turbines of the few-stage type like the Curtis. The area of the nozzles between the second and third stages, and often also between the first and second stages, is invariable with the load if there is no overload by-pass valve. The pressure drop is always great enough when steam is supplied at boiler pressure to make Napier's formula (page 189) for the flow of steam applicable; that is, the weight of dry saturated steam passing through these nozzles of constant area is proportional to the pressure on the "inlet" side. If the area of the nozzles in one of these stages is known where the steam is approximately dry and saturated, and an ordinary recording pressure gage is attached to indicate the pressure on the "inlet" side of the nozzles, the weight of steam can be determined much more accurately than with any of the other automatic devices yet devised. If desired the chart of the recording gage can be readily graduated by a skillful draftsman to indicate directly pounds of dry steam. If the steam is superheated a correction curve can be readily made by applying the formula (55).

Float Steam Meters are designed so that a float, usually a disk or a cone, moves against a constant resistance in a passage in which the unrestricted area for the flow of steam varies with the height of the float. This principle is applied in the St. John and Sargent meters in each of which there is a conical float connected to the registering device showing the rate of flow. In the St. John meter (**Fig. 237**) the float V rises with increased flow, carrying with it the arm N connected to the registering device. Accuracy of steam meters is usually not greater than ± 5 per cent.

Prices of steam meters: "G. E." Pitot tube \$85 to \$165 for indicating types, \$260 to \$270 for recording types (General Electric Co., Schenectady, N. Y.); St. John Recorder \$250 for 2-inch pipe, \$550 for 6-inch pipe (G. C. St. John, New York); Sargent \$200 for 2-inch pipe, \$450 for 6-inch pipe (Pittsburg Supply Co., Pittsburg, Pa.).

FIG. 237.—St. John Steam Meter.

The Flow of Water. When the quantity of water to be measured is not too large it is most accurately determined by weighing in tanks placed on scales, or by direct measurement of volume in calibrated tanks or barrels. Sometimes it is impracticable to weigh or measure the

volume of the water directly, particularly when it must be measured **under pressure.** For measurements in pipes up to 2 or 3 inches in diameter a **water meter** is generally used.

A great many types of water meters are sold commercially and not very many are accurate, so that it is absolutely necessary to calibrate them at least before and after a test, under the same conditions of temperature, pressure, and rate of flow. In many plants where meters are used constantly, suitable connections are made to the discharge from the meter, so that at any time the flow through it can be diverted into a tank in which it can be measured by volume or weighed.

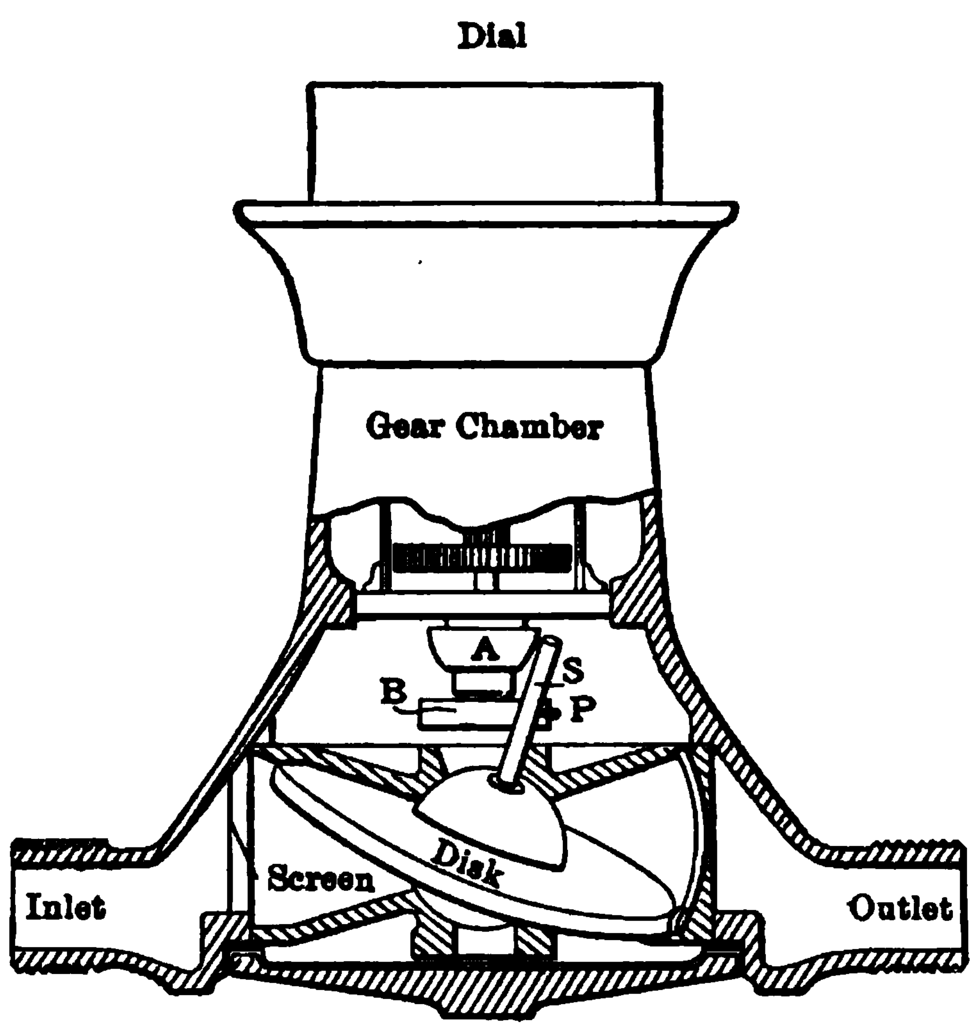


FIG. 238. — Pulsating Diaphragm Water Meter.

One of the best types of water meters is illustrated in **Fig. 238.** This belongs to the class operating with a "pulsating diaphragm." The inclined shaft S on this diaphragm traveling around in contact with the peg P on the plate B moves the counting mechanism through intermediate gears. This diaphragm in the **Thomson-Lambert meter (Fig. 239)** is made of hard rubber reinforced with a steel plate, making it much more durable than those made without reinforcing. As the side chambers are alternately filled and emptied, the diaphragm is moved up and down with a kind of "pulsating" motion and operates the recording mechanism. The diaphragm divides the measuring chamber into two compartments of equal volume. While one of these is filling the other is emptying. For general purposes this type is probably used more than any other, and for a fairly constant flow is quite accurate; but because of leakage it is not accurate for rates of flow that at times have very low values.

Piston Water Meter. The piston type of water meter **(Fig. 240)** is also used frequently. It belongs to the type operating in a cylinder by a reciprocating piston which is driven backward and forward by the pressure of the water. In this device there are two pistons side by side. Water is admitted alternately at each end by a slide valve **A** moving on seats in the plate **S,** just above the bottom casting containing the inlet and outlet chambers. These valves are moved by

Fig. 239. — Thomson-Lambert Water Meter.

Fig. 240. — Piston Water Meter.

contact with the inner faces of the plunger heads near the end of the travel and move them over at the proper time. The lever L is moved back and forth by one of the plungers to operate the counting mechanism. The cored passages in the bottom casting are too complicated to

be shown clearly. The plungers at the end of their travel strike against the rubber bumpers **R**, which are provided to reduce the shock.

Meters actuated by the **velocity** of water are particularly suitable for measuring large quantities at low pressure. **Fig. 241** shows an example of this class. Water flows into the wheel **I** after entering and passing through the screen **S** as shown by the arrows. Guide vanes deflect the water horizontally and radially outward from the center into the discharge passage **A**. The wheel when it revolves moves the counting mechanism **G** above. Objections to such meters are that they are very unreliable for small flows because of the friction of the parts, and an appreciable flow is required to start them. Friction is an important element in meters of this type, but they are not injured by moderately hot water.

FIG. 241. — Water Meter Operated by Velocity.

The readings of a water meter are usually in cubic feet. A water meter is essentially a water motor adapted for operating the gearing connected to the counting mechanism.

Frequent calibrations of water meters are necessary because they are likely to become more or less clogged with dirt and refuse. The readings are also affected by the temperature, head, and quantity of water flowing, as well as by the amount of air carried in the water. A meter should always be calibrated at least at two or three rates of flow, as it scarcely ever happens that the conditions of the test are so uniform that the meter will be used only for a certain predetermined rate of flow.[1]

Willcox Water Meter. Automatic measuring devices are often used for determining the weight of condensed steam in engine tests or the weight of feed water in boiler trials. The Willcox meter[2] is a most satisfactory apparatus of this kind. It consists of a tank (**Fig. 242**) divided by a partition **P** into two compartments **A** and **B**, one above the other. The upper compartment **A** receives the inflow of water and the lower one **B** serves for measuring. Projecting into the lower compartment is a U-shaped discharge pipe **C**, which is always water-sealed. The upper end of the discharge pipe is covered by a bell float **F**, which is permitted a short up-and-down movement. In the upper compartment there is a

[1] Calibration curves are usually plotted with meter readings as abscissas and actual volumes as ordinates. A curve should be plotted for each of the several rates of flow if they are different. Curves of meter readings (abscissas) and correction factors (ordinates) are also useful.

[2] Willcox Engineering Co., Saginaw, Mich.

short standpipe S, which is simply a hollow cylinder open at the top and bottom. The bell float F and the standpipe S are connected rigidly by a vertical rod (**Fig. 243**) so that they move together as one piece, and this is the only moving part in the apparatus. The lower end of this standpipe has a corrugated face, and when it is down in its lowest position its corrugated face rests on a soft seat or ring surrounding a circular opening in the partition P. This seat is made of a rubber composition which is not injured by boiling water. The apparatus can be used, therefore, with either hot or cold water without risk.

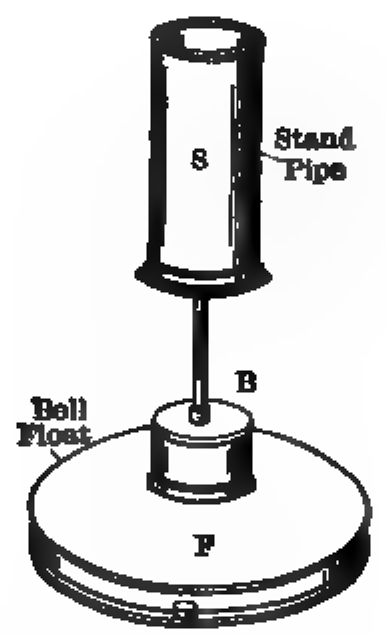


FIG. 242. — Willcox Water Weigher (by Volume). FIG. 243. — Bell Float.

In the operation of the apparatus, when the standpipe S is down on its seat, water entering through the side inlet accumulates in the upper compartment A until it overflows the top of the standpipe. The water then flows down through the hollow standpipe into the lower compartment until there is a sufficient amount to seal the lower edge of the bell float F. Then as more water accumulates the bell float rises, lifting the standpipe S from its seat and the water in the upper compartment flows down into the lower one until the volume is that of a "unit charge" for the apparatus, when the "tripping" device discharges the water through the discharge pipe C. The "tripping" is accomplished by a "trip" pipe T, which is normally water-sealed, but which becomes unsealed when a "unit charge" has accumulated. While the water is accumulating in the lower compartment B the water in the left-hand leg of the "trip" pipe T is being slowly pushed down

because of the increasing pressure of the air under the bell float **F**, due to the increase of head of water, and a corresponding amount of water spills over the upper end of the right-hand leg **R** of the " trip " pipe into the discharge pipe **C**. Due to this action the water level in **T** is lowered until it reaches the bend in the lower end of the " trip " pipe. Under these conditions the water column in **R** exactly balances the head of water in the lower compartment **B** and the air entrapped in the float valve **F** has a function similar to that of a scale beam, balancing on one side the head of water in the tank and on the other side the head of the standard water-column in **R**, which of course is always constant. At the instant this balance is secured a very small amount of water added in the lower compartment and the corresponding additional spill from **R** will destroy this equilibrium. Then the air compressed in the bell float **F** and the upper part of the discharge pipe breaks the water seal in **R** by suddenly discharging all the water in it. When the air pressure in the bell float **F** is thus reduced it drops down, carrying down with it the standpipe S in the upper compartment **A**. In this last operation the air is moved from the interior of the bell float **F**, and water flowing in to replace it will spill over the top of the discharge pipe **C** and will flow out at the other end until the lower chamber is emptied of the " unit charge." At this time the standpipe **S** becomes seated, due to the pressure of the water above the bell float and to the downward suction of the syphon. Thus the standpipe is held tightly upon its seat only at the instant when tightness is required; that is, while the " unit charge " is being discharged. After the standpipe has seated water again accumulates in the upper compartment **A** and the cycle of operations is repeated.

A mechanical counter shown at the side of the apparatus is connected to a ball float inside the lower compartment and registers the number of times the apparatus is tripped. An automatic device of this kind is easily calibrated by weighing several " unit charges," and it can then be used with as great a degree of accuracy as can be expected with rapid weighings in tanks on platform scales. It may be expected to weigh hot or cold water with a maximum error of not more than one per cent for the conditions of calibration.

Leinert (Worthington) Weigher. The apparatus shown in **Fig. 244** is one of the few devices made which actually weighs the liquid to be measured. It can therefore be used for liquids of widely varying density without making corrections. It consists of a pair of open tanks supported on trunnions with knife-edge bearings (**K**) in such a way that when empty they assume the normal position as shown in the figure. After a certain quantity of liquid has entered the tank the counterbalancing effect of the lead weights in the casing **W** is overcome and the

tank tips over and the contents are siphoned out through the pipe **P.** By this tipping action the trough **H** receiving the liquid from the supply pipe I is switched from the full to the empty tank, and at the same time the counter **C** is operated to register the number of times the tanks have been filled. Since weight is the method of measurement the record is independent of the temperature of the liquid; but there is always a very small variation of weight with the rate of flow. The weight per charge can be adjusted by changing the number of weights in the casing **W**[1].

Fig. 244. — Tilting Tank Water Weigher.

Venturi Meter. An arrangement of piping in which there is a gradual narrowing of the section to a minimum followed by a more gradual enlargement was invented by Mr. Clemens Herschel for measuring the flow of water. This apparatus is called a venturi meter and is shown in **Fig. 245.** Piesometer tubes (manometers) are arranged to indicate the pressure at the sections shown. Pressures at these sections will be denoted respectively by p_m and p_n.

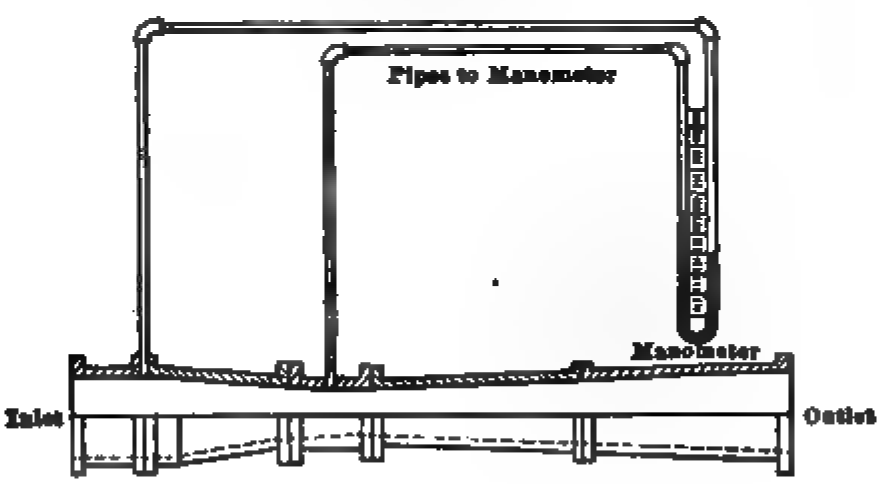


Fig. 245. — Herschel's Venturi Meter.

From Bernouilli's theorem[2] it follows that the relation between the pressure in pounds per square foot and the velocities in feet per second at the two sections v_m and v_n of a stream flowing through such a closed horizontal channel is given by

$$\frac{p_m}{\delta}+\frac{v_m^2}{2g}=\frac{p_n}{\delta}+\frac{v_n^2}{2g}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

[1] Henry R. Worthington, 115 Broadway, New York, and Holden & Brooke, London.

[2] See Jamieson's *Applied Mechanics*, vol. 2, page 458.

where δ is the density of the water in pounds per cubic foot. If a represents the area of a section in square feet, the volume of water flowing through any section, (cubic feet per second), is

$$a_m v_m = a_n v_n = a_n \sqrt{\frac{2\,g\,(p_m - p_n)}{\delta\left(1 - \frac{a_n^2}{a_m^2}\right)}} \quad . \quad . \quad . \quad . \quad . \quad (57)$$

With suitable manometers or with gages the pressures p_m and p_n can be obtained, and since all other quantities can be represented by a constant k, we have

$$\text{Volume per Unit of Time} = k\,(p_m - p_n). \quad . \quad . \quad . \quad . \quad . \quad (58)$$

As usually made the venturi tube is merely a pipe which tapers from each end towards the throat, which is usually lined with hard bronze to secure a smooth bore and has a diameter of from $\frac{1}{3}$ to $\frac{1}{2}$ that of the pipe line. Its total length is about eight times the diameter of the pipe. In the commercial forms of this apparatus near the inlet or up-stream end and also at the throat are annular chambers encircling the tube, which communicate with the interior by numerous small vent holes. When no water is flowing in the venturi tube the pressure will be the same in these two annular chambers; but when there is a flow of water through it the throat pressure becomes less than the up-stream pressure. The difference between the two is proportional to the square of the velocity of the water.

A recording device[1] has been arranged, consisting essentially of a large U-tube serving the same function as the one in the figure filled with mercury, supporting in each leg an iron float. These floats have toothed racks connected to their upper ends which engage with pinions on the same horizontal shaft with a cam. A small wheel supporting the recording pencil rides on the perimeter of this cam, which is arranged so that the wheel rides on the greatest eccentricity of the cam and consequently the recording pencil will indicate on the chart the greatest flow when the rack is at its maximum height. The pipes connecting the U-tube with the venturi meter should always be full of water. Air cocks are provided for removing air that may accumulate in pockets.[2] Since the flow is proportional to the square root of the difference of pressure a complicated cam device is required to permit the charts of the recorder to be made with equal divisions.

The General Electric Flow Meter (see page 191) is also sometimes applied for measuring the flow of water in pipes.

[1] Builder's Iron Foundry, Providence, R. I.

[2] For more detailed discussion and tests see Herschel's papers in *Trans. American Society of Civil Engineers*, Nov., 1887 and Jan. 1888; also *Power*, Jan. 23, 1912.

Flow of Water through Orifices and Nozzles. Theoretically the velocity of flowing water under any pressure is the same as the velocity attained by a body falling freely through a distance equal to that head (**h**) as in **Fig. 247.** Furthermore this statement would be the same even if the water had no free surface, provided, however, the **pressure** at the orifice was that due to a head **h.** If then there is no loss of head due to friction and eddies formed by the water passing through the orifice the velocity of discharge, **v** in feet per second, is

$$\mathbf{v} = \sqrt{2\,\mathbf{gh}}, \qquad (59)$$

where **g**[1] is the acceleration due to gravity and **h** is the head over the center of the orifice in feet.

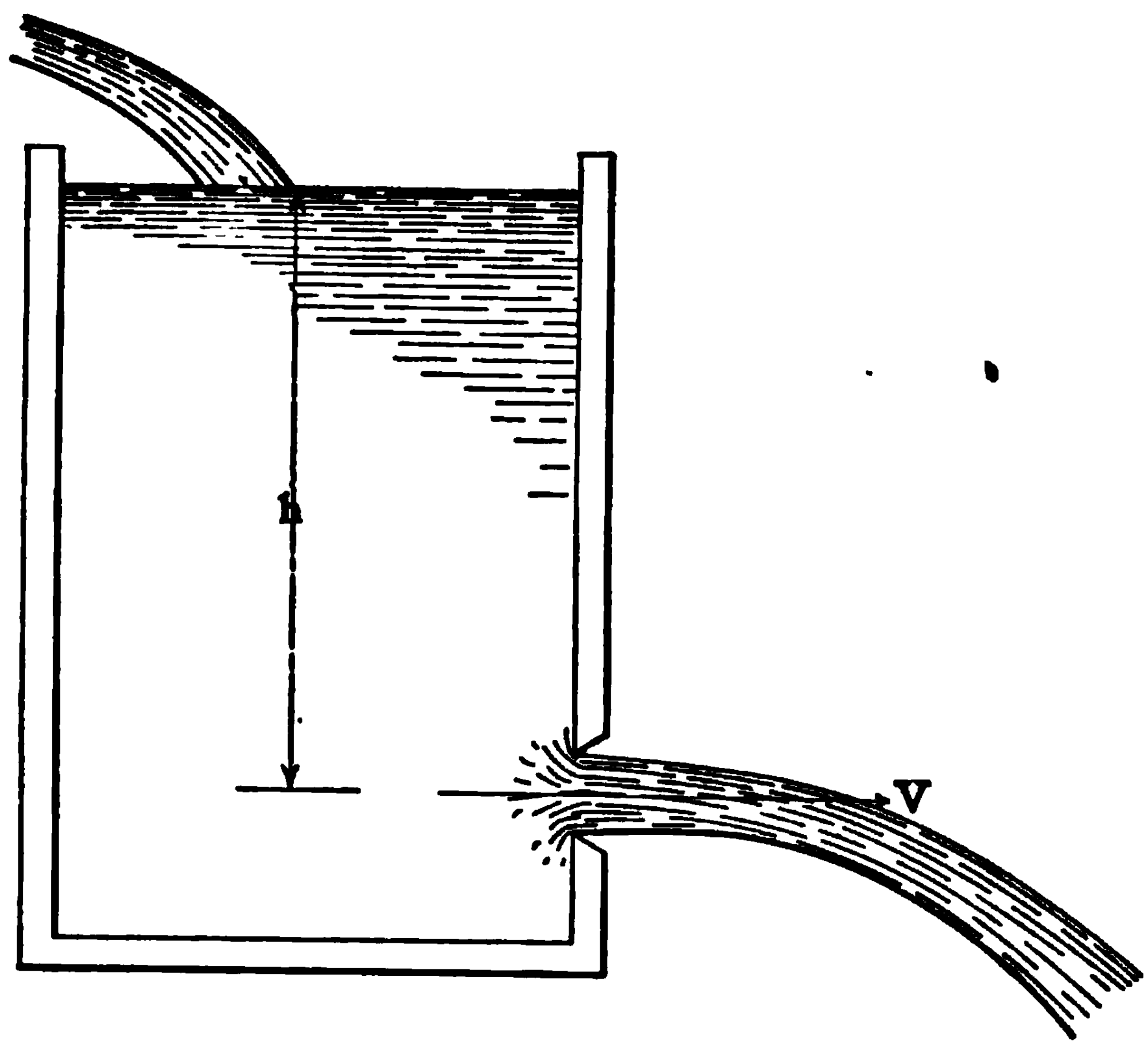


FIG. 247. — Discharge of Water from an Orifice.

If **a** is the area of the cross-section of the orifice in **square feet, q** is the quantity or volume of water discharged in **cubic feet per second,** and assuming the stream is of the same cross-sectional area as the orifice, then

$$\mathbf{q} = \mathbf{a}\sqrt{2\,\mathbf{gh}}. \qquad (60)$$

Since the actual flow is less than the theoretical in most cases, and considerably less when the discharge is from a hole with sharp edges in a thin plate, a more general form may be written by inserting a suitable coefficient[2] of discharge **k,** then,

$$\mathbf{q} = \mathbf{ka}\sqrt{2\,\mathbf{gh}}. \qquad (61)$$

[1] The value of **g** is approximately 32.2, so that equation (59) can be simplified into $\mathbf{v} = 8.02\sqrt{\mathbf{h}}$.

[2] This coefficient is often called the coefficient of contraction.

For an orifice located in the side or bottom of a tank, consisting of a circular opening in a thin metal plate with a smooth **sharp** edge, the value of the coefficient **k** may be taken as 0.6 for all practical purposes. (See Report Power Test Committee of A.S.M.E. in *Journal*, Nov., 1912, page 1829, and Hamilton Smith, Jr's, *Hydraulics*.)

Calibration of Orifices and Nozzles. Water under a constant pressure is often measured by observations of the flow through either orifices or short nozzles which have been carefully calibrated. The apparatus required for this calibration consists usually of a suitably arranged standpipe to which the orifice or nozzle can be attached so that a given head of water can be maintained[1] and barrels on scales (or a tank calibrated for volumes) to receive the water discharged.

A pressure of one pound per square inch is equivalent to a head of water at 62 degrees Fahrenheit of 27.72 inches, or 2.31 feet. **A normal atmospheric pressure** (14.7 pounds per square inch) is therefore equivalent to a **head of 33.96 feet of water.** Then for a given pressure or head the quantity of water discharged in a given time is readily obtained and the coefficient of discharge can be computed by substituting the values of quantity of discharge **q**, the head **h**, and the area **a**, in formula (61).

Data and results should be tabulated in the form given below. The relative roughness of the edge of the orifice or of the inside surface of the nozzle should be recorded.

FLOW OF WATER

Flow of water through a..

Date..............................Observers..............................

Form of orifice or nozzle............Formula..........Diameter, feet...........

(Sketch) ....Area, square feet

No. of Reading.	Head in Feet.	Time in Seconds.	Total Pounds or Cu. Feet.	Pounds per Second.	Cu. Feet per Second.	Coefficient of Discharge (k).	Remarks.
Average							

[1] In many places a suitable pressure tank is not available, and in such cases the calibration can be made by attaching the orifices or nozzles to pipes carrying water under pressure. The readings of the pressure gage can be reduced to the equivalent head in feet, to which must be added, if the center of the gage is higher than the orifice or nozzle, the distance in feet from the center of the nozzle to the center of the gage.

Curves. Curves should be plotted for each orifice or nozzle with head in feet for abscissas and (1) the discharge (cubic feet per second) and (2) the coefficient of discharge for ordinates.

Flow of Water over Weirs. When large quantities of water are to be measured, then orifices are unsuitable and it is customary to pass the whole body of water over a **weir** or **gage notch.** This consists of a board placed across the stream so that all the water must pass over it. The length of the notch is usually made less than the width of the stream to give definite conditions. This is accomplished most easily by sawing the notch out of a long board and beveling the edges.

A typical arrangement for measuring the head of water on a weir is illustrated in **Fig. 248.** The head must be determined with great accuracy, and this is done usually by means of a **hook-gage, Fig. 249,** and a suitable machinist's or carpenter's level.

FIG. 248. — A Weir for Measuring Water.

FIG. 249. — A Hook Gage.

The hook-gage consists of a sharp-pointed hook **H,** attached to a vernier scale **V,** intended to measure very accurately the amount the hook is moved. Before taking an observation the hook must first be submerged and then raised slowly till the point just breaks the surface of the water. The correct height of the surface is obtained at the instant when the point of the hook pierces it. The head **h** of the water flowing over the weir (**Fig. 248**) is obtained by setting by means of a straight-edge **SE** and the level **L** the point of the hook at the same level as the crest of the weir. The height observed in this position is called the **zero head.** It is to be subtracted from all other readings to get the head of water flowing. The hook-gage must be placed in such a position on the upstream side of the weir where the surface has no

appreciable velocity and where there is very little disturbance due to eddies. In terms of the following symbols,

q = quantity or volume of water discharged in cubic feet per second;
h = the head in feet on weir measured in still water;
b = breadth of the weir in feet;
n = the number of contractions;
k = coefficient of discharge.

$$q = 2/3\, kh^{3/2}(b - 0.1\, nh)\sqrt{2g}. \qquad (62)$$

This is the well-known Francis formula for a rectangular notch. The ordinary rectangular notch has two contractions, one at each side of the crest. **Triangular notches** in weirs are sometimes used. One of these in the form of a right-angled isosceles triangle is shown in **Fig. 250.** It has the advantage of giving the same form of stream what-

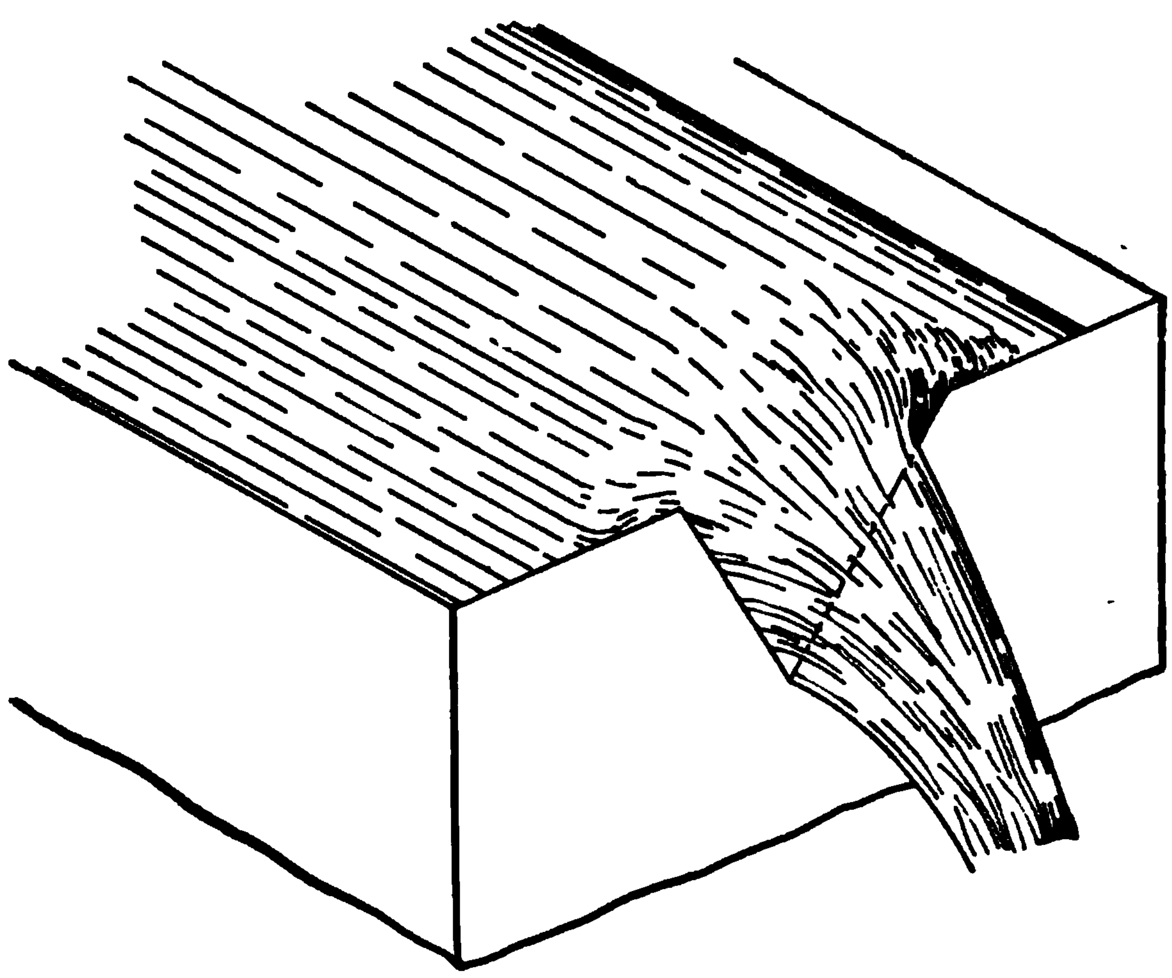


FIG. 250. — Weir with a Triangular Notch.

ever the size of the notch or the height of water passing through. It is, therefore, particularly suitable for measuring a flow of water which is somewhat variable The quantity of water discharged over a triangular weir or notch is

$$q = 4/15\, kbh^{3/2}\sqrt{2g}. \qquad (63)$$

When the angle is 90 degrees,

$$b = 2\,h \quad \text{and} \quad q = 4.26\, kh^{5/2}. \qquad (64)$$

Also when the angle is 60 degrees,

$$b = 2\,h \tan 30° \text{ and } \quad q = 2.47\, kh^{5/2}. \qquad (65)$$

3. Weir Meters. A weir or notch will measure any quantity of water if made of a suitable size. Rectangular weirs are generally used for large quantities and notches of various shapes for small quantities. For accurate work the head on the crest is measured with a hook-gage but in many cases a float is used. The instrument used for measuring the head should not be less than two feet from the crest and preferably farther away. The distance of the crest from the bottom of the weir tank should be not less than three times the average head. For a weir with two contractions the width of the tank should be not less than three times the width of the weir. To maintain the surface free from ripples baffle plates, preferably of perforated sheet metal, should be located between the supply pipe and the instrument for measuring the head.

Lea's Recorder[1] **for V-notch Weirs** is successfully used in many services for measuring water. Water level is measured by a sheet-metal float usually about 12 inches in diameter which is attached to the vertical shaft S of the recorder in **Fig. 251.** On the upper end of this shaft is a rack R meshing with a pinion on the left-hand end of the horizontal shaft carrying the drum D which has on its curved surface a spiral band over which a trailing-arm F is fitted. The curvature of the spiral band has been made to conform to a logarithmic curve so that increments of movement of the trailing-arm are proportional to the quantity of water flowing, and not to the up and down movement of the float. The trailer F is connected to the pen-arm P, making a record on the paper drum C. These charts will therefore have equally spaced ordinates and the area under the curve traced on them is proportional to the quantity of water flowing in a given time.

Fig. 251. — Lea's Recorder.

Moyer's Recorder (Fig. 252) consists of a weir tank T into which water discharges through the supply pipe S. A float F is located under the recorder R. The vertical shaft of the float, held in line by small ball-bearing guides, is connected to the pen point P directly, without any intervening gears or linkages. Baffle plates B are placed between the supply-pipe and the float to eliminate ripples and steady the float. The notch N follows almost exactly the theoretical lines for a flow pro-

[1] Yarnall-Waring Co., Chestnut Hill, Pa.

portional to the head, accurate allowances being made also for end contractions. Radial ordinates traced by the pen point **P** are therefore proportional to the rate of flow. By measuring these charts with a Bristol-Durland averager for circular charts (see page 87) the quantity of water flowing in a given time can be accurately obtained.

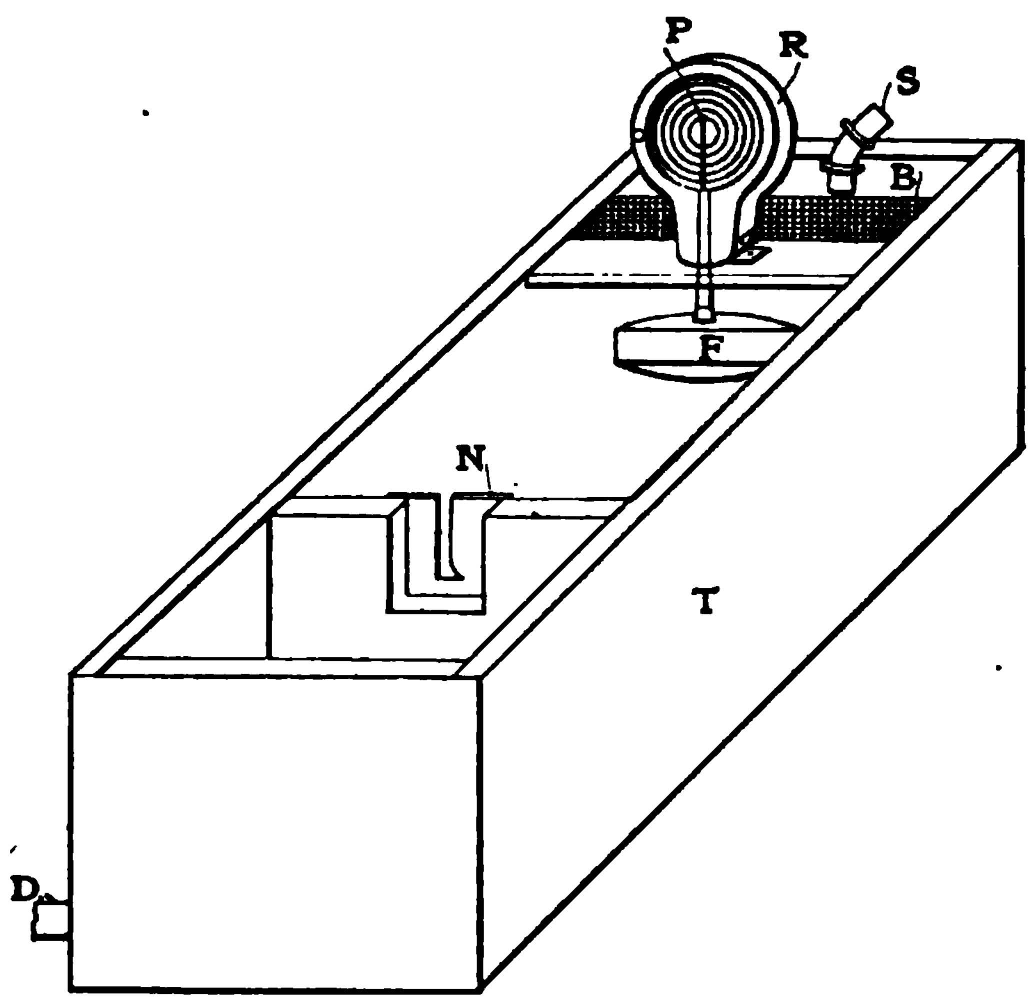


FIG. 252. — Moyer's Recording Weir.

The charts are graduated in pounds of water per hour instead of cubic feet. This is made possible by the automatic temperature correction of this apparatus. For a given weight of flow as the temperature increases or decreases the head increases or decreases correspondingly and **vice versa;** but at the same time the displacement of the float increases as the water becomes lighter by reason of being hotter. These two influences therefore counterbalance, making the weight discharged in a given time proportional to the ordinates of the curve traced by the pen of the recorder. This apparatus is accurate to ½ per cent for temperature correction.

An automatic float valve is provided for shutting off the water supply to prevent overflowing.

Lea's Recorder indicates also pounds per hour, but because the movement of the float is not proportional to the head the automatic correction for temperature is only an approximation.

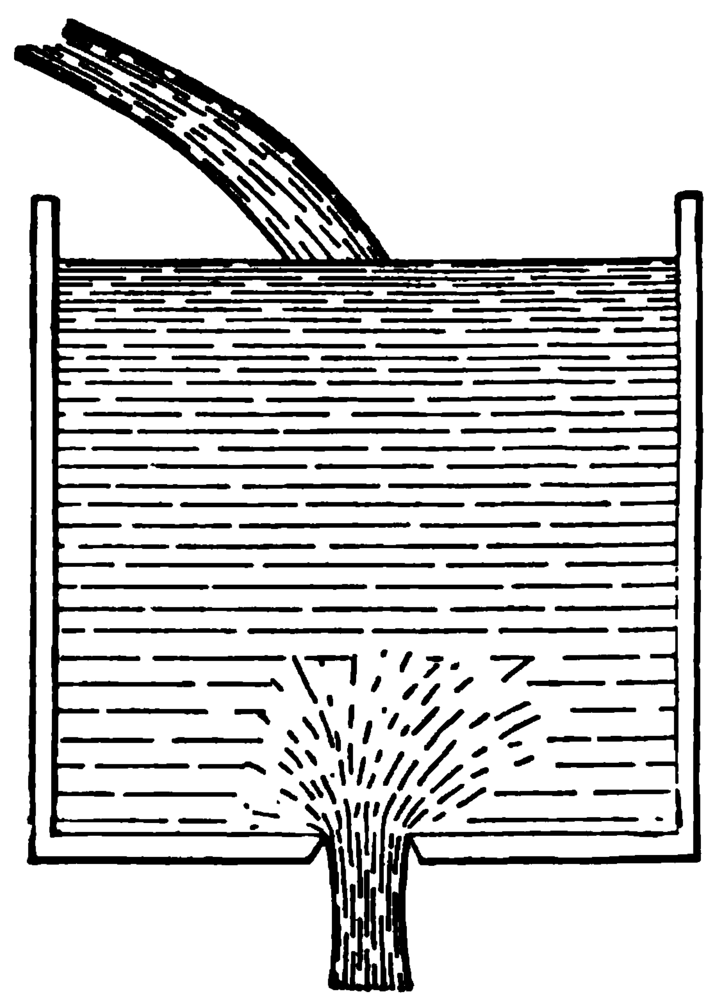


FIG. 253. — Best Kind of Orifice for Engine Tests.

No weir device is very accurate for very low rates of flow. Any mistake made in determining **h** will produce a larger percentage error in the results with the rectangular and triangular notches than with an orifice. Where great accuracy is desired and the quantity of water to be handled is not too large, an orifice calibrated

and used in the bottom of the tank as shown in **Fig. 253** is to be preferred to measurements with a weir. This remark is particularly applicable in connection with the measurements of cooling (circulating) water in tests of large steam engines and turbines.

Calibration Data Sheets. Use the same form for data as given for calibration of orifices or nozzles on page 202.

Curves should be plotted with heads for abscissas and (1) the discharge (cubic feet per second) and (2) the coefficient of discharge for ordinates.

Weighing Liquids in Tanks. In order to weigh liquids under a continuous flow two tanks, usually made of sheet metal, are generally used. Each tank is placed on a scales and the liquid is alternately discharged into each. The discharge pipe is usually arranged so that it is movable from one tank to the other by turning on a screw thread. At the side of each tank near the bottom a so-called "quick opening" valve or cock is provided for rapidly emptying the tanks. These discharge valves should be large, because the more rapidly the tanks can be emptied the greater the quantity of water the arrangement will handle.[1]

When only one platform scales is available an arrangement like that shown in **Fig. 254** can be used efficiently. The larger tank is placed on the scales and the smaller one is supported on a platform or bench at such a height that it can be readily discharged into the larger tank. During the operation of weighing and emptying the larger tank, the liquid is discharging into the smaller one; and when the discharge is again directed into the larger tank the valve or cock on the smaller one is opened so that its contents will be included in the next weighing of the larger tank.

For weighing feed-water in boiler tests the reverse of this arrangement is frequently applied. There are as before two tanks or barrels, of these the one more elevated is on a platform scales, and the attendant doing the weighing empties weighed quantities of water into the lower tank as needed to supply the feed-pump. In the lower tank the water level must be the same level at the end as at the beginning of a test.[2]

In cases where the flow is absolutely constant as in the discharge of water from nozzles or orifices with a constant head a tank may be filled at intervals, observing accurately the time for filling with a stop-watch and weighing each time. The average of several such determinations gives a fairly accurate result.

[1] The discharge can be increased by attaching a short pipe to the discharge side of the valve which by reducing the contraction increases the flow.

[2] For determining these levels in the tank a water-gage glass is very convenient. If, however, there is no gage glass on the tank marks can be made with a knife-scratch or by painting a line on the inside of the tank.

Liquids are also often measured instead of weighed in calibrated tanks. In every case the temperature of the liquid must then be observed. In some cases the tanks have graduated scales at the side of a glass water gage from which the volume of water can be observed; or again there is only a single mark up to which the tank is to be filled each time. Establishing the exact level for a large surface is not an easy matter and to make this method more accurate the marks up to which the tank is to be filled are preferably put on a portion of the tank at the top which has been made considerably smaller in size than the rest of the tank.

It is a very poor method to fill tanks up to the rim on account of the variableness of the meniscus which may vary from various causes.

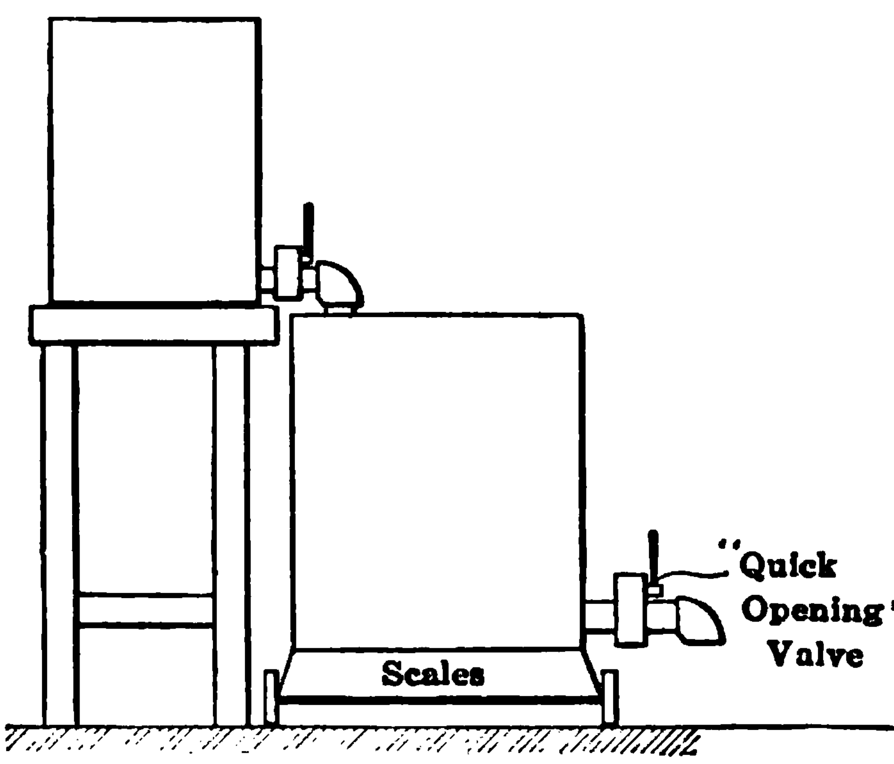


FIG. 254. — Weighing Device.

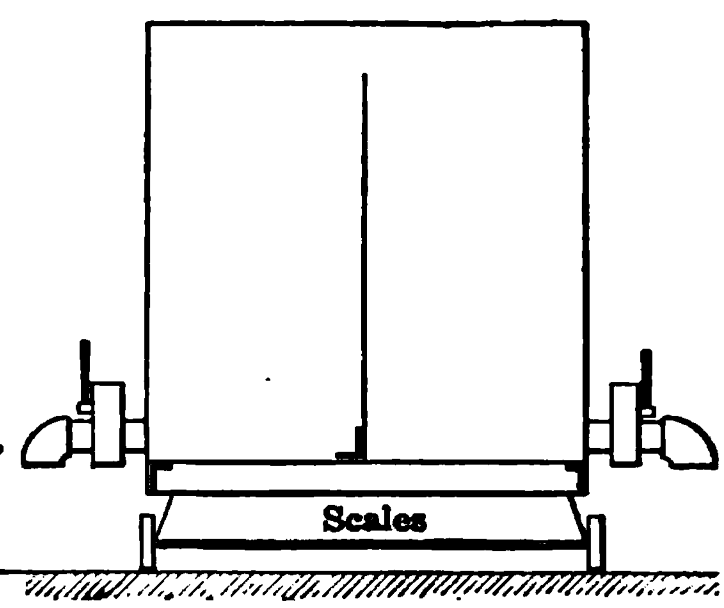


FIG. 255. — "Double" Tank.

Actual weights corresponding to measurements of volume are always varying with the temperature of both the liquid and of the tank or collecting vessel. It is therefore necessary in every case to determine by actual tests the weight corresponding to the volume of a tank for a given liquid at various temperatures and apply a calibration curve for temperature variations to all measurements. Calibrations should be made also with the inside wetted surface in as nearly the same condition as it will be after each emptying in a test.

An ingenious method of measuring the flow of a liquid is illustrated in **Fig. 255**. It shows a tank separated by a partition into two parts. The partition is not quite as high as the sides of the tank. In operation one of the halves is filled up to the top of the partition, permitting any excess to flow over into the other half. Then the supply pipe is swung over to the second half which is filling while the first half is being emptied. The tanks must be emptied, of course, more rapidly than they are filled.

In order to make easier the supervision and checking of tests it is desirable that as nearly as possible equal quantities should be weighed or measured as the case may be.

Automatic scales provided with a continuously operating controlling device have been successfully developed. The poise on the weighing beam is set in motion when the article to be weighed is put on the scales and when the beam shows it is balanced it stops automatically. The weight is usually registered by means of a counting device but a printing recorder can also be used.

CHAPTER VIII

CALORIFIC VALUE OF FUELS—SOLID, LIQUID AND GAS

Calorific power is a term applied to the quantity of heat generated by the complete combustion of a definite quantity of fuel. In order to insure rapid and complete combustion the fuel is **preferably** burned in an atmosphere of oxygen under pressure. This calorific power of fuels is expressed in the English system as **British thermal units per pound,** and in the metric system as **calories per kilogram.**

In fuel calorimetry it is always assumed in engineering calculations that at about the usual " room " temperatures the specific heat of water is constant, so that the weight of reasonably pure or distilled water in pounds times the change in temperature in degrees Fahrenheit is the heat change in British thermal units (B.t.u.). Similarly the weight of water in kilograms times the change of temperature in Centigrade degrees is the heat change in kilogram-calories (French) or Wärme Einheiten (German). For conversion constants see page 29.

The quantity of heat generated by combustion is measured by the rise in temperature of a given weight of water in a calorimeter of which the cooling effect or **water equivalent k** has been determined, and the temperature of any gas escaping has been reduced to that of the room. Now if

$\mathbf{w}_f$ = weight of the fuel in pounds,
$\mathbf{w}_w$ = weight of the water in pounds,
$\mathbf{k}$ = water equivalent[1] of the calorimeter, in pounds,
$\mathbf{t}_1$ = initial temperature of water, degrees Fahrenheit,
$\mathbf{t}_2$ = final temperature of water, degrees Fahrenheit,
$\mathbf{Q}$ = total heat generated, B.t.u.

[1] Water equivalent is used to express the heat-absorbing effect of the calorimeter as equivalent to that of a weight of water. This may be found (1) as for calorimeters used for determining the quality of steam by the hot-water method (see page 72) (2) by taking the sum of the products of the weights and specific heats of the various parts of the calorimeter (see *Calorific Power of Fuels*, by H. Poole, pages 14 and 15), or (3) by comparing the results obtained with those that should have been secured, if there had been no absorption of heat, by the combustion in oxygen gas of some substance of which the heat value is known; as, for example, pure sucrose, carbon, napthalene, or benzoic acid. Samples of the materials of which the heat value has been worked out very accurately may be obtained together with a certificate at a very small expense from the U. S. Bureau of Standards, Washington, D. C. The standard sample

then the calorific value **H** per pound of fuel in British thermal units is

$$\mathbf{H} = \frac{Q}{w_f} = \frac{(w_w + k)(t_2 - t_1)}{w_f} \quad \ldots \ldots \quad (68)$$

Corrections for Radiation can be practically eliminated by having the temperature of the water in the calorimeter before ignition as much below the " room " temperature as the final temperature is above.

Bomb Calorimeters. Formerly the calorimeters used for burning fuels in an atmosphere of oxygen were arranged for combustion at **constant pressure,** but since it was found that more reliable results could be obtained generally with apparatus maintaining a **constant volume,** the former type is not now much used. When the combustion takes place at **constant volume,** the vessel receiving the charge of fuel and oxygen must be designed to withstand a great pressure, and therefore on account of the massive construction required the vessel is called a **bomb calorimeter.** The essential part of such a calorimeter is the strong steel vessel or bomb similar to **Fig. 256.** It consists essentially of a steel shell **S** having a capacity of about 50 cubic inches and capable of resisting with safety a pressure of about 750 pounds per square inch. This shell is usually provided with a coat of enamel or a lining of platinum or nickel on the inside and is nickel-plated on the outside. The coating or lining on the inside is intended to resist corrosion and oxidizing action during the combustion. The advantage of the nickel lining over the coat of enamel is that when it is worn out or broken it can readily be replaced and at much less expense than the enamel. The shell is closed at the top by an iron cover or cap which is to be made tight by screwing down on a lead washer with considerable force, using a long wrench. At the top of this cover or cap there is a conical seated valve, which is screwed in through the gland and stuffing-box **G,** by attaching a wrench at **P.** The valve and its seat are made of good nickel, as this metal is not easily oxidized. A wire electrode **B,** which is well insulated from the cover, extends into the shell and con-

FIG. 256. — Section of Bomb Calorimeter.

should be made into a pellet (see page 216) weighing about 1.5 grams which immediately after weighing should be put into the calorimeter and burned in commercially pure oxygen gas at about 400 pounds per square inch. After correction for radiation (see page 214) the discrepancy in the heat balance is the product of the required water equivalent (pounds) and the observed temperature range. Detailed directions for very accurate determinations are given in Circular No. 11, of U. S. Bureau of Standards.

ducts the electric current for firing the charge of fuel, which is placed on a platinum dish or crucible supported by another wire **A**, attached to the cover on the inside.

Usually one gram of finely powdered coal which will pass through a sieve having 100 meshes to the inch (" 10,000 meshes to the square inch ") is put into the dish to make a test for calorific value.[1] A small iron wire (which was previously weighed) is then suspended over the dish between the electrode and the wire support for the dish. The cover should then be screwed on with a long wrench, the shell itself being held in a vise. The **complete Mahler apparatus**[2] is shown in **Fig. 257**, showing the cylinder of oxygen **O**, the pressure gage **M**, the calorimeter vessel **D**. The end of the conical-seated valve (**Fig. 256**) is

FIG. 257. — Complete Mahler Apparatus.

attached by means of pipe connections, preferably flexible, to the union **U** and to the valve **W**, which because of the high pressure should be opened slowly and carefully, and allow sufficient oxygen to pass into the bomb to provide a considerable excess above that actually required.

[1] The Power Test Committee of A.S.M.E. recommended that the sample should be " air dried" (see page 232); and that a similar "air dried" sample should be tested for moisture, so that the final result may be based on heat value per pound dry coal. Many chemists and engineers prefer to use a sample of coal powdered "as received" and determine the heat value per pound "as received." The latter method does not give as accurate results as the former, but slightly reduces the time required for making the calorific determination.

[2] The Mahler type of calorimeter is recognized as the most complete and accurate apparatus of its kind. Where the engineer does not have this instrument or some other reliable calorimeter of similar construction, the heat units can be determined by sending samples to a testing laboratory where such instruments are used." — *Report of Power Test Committee of A.S.M.E. in Journal*, Nov., 1912, page 1698.

The pipes for connecting the bomb to the oxygen cylinder should connect also with a pressure gage as shown, so that the pressure in the bomb can be regulated. For the combustion of coal a sufficient volume of oxygen is admitted to Mahler bombs of the usual size to make the pressure in the bomb from 350 to 375 pounds per square inch. Now close the valve on the oxygen cylinder and the conical-seated valve on the bomb, removing also the connections between the bomb and the oxygen cylinder. Oxygen should be admitted to the bomb slowly, because if accidentally the oxygen be allowed to go in a little too rapidly, some of the sample of coal will be blown out of the dish and will probably not be burned. The bomb should then be placed in the calorimeter vessel **D,** which should be filled with a quantity of water previously weighed (at least about five pounds) to fill it to about the level indicated in the figure. Place the calorimeter thermometer **T** into the vessel, being careful that the end will not be touched and broken by the stirrer or other parts, and then after agitating the water for a few minutes to establish a uniform temperature, the observations can begin. The temperature should be very carefully observed for five minutes and recorded **minute by minute,** to determine the rate of variation of temperature before combustion. Then the electric circuit should be made and the combustion will, of course, begin immediately; but some little time will be required for the transmission of the heat generated to this water. Now take the temperature at the end of a half minute after making the electric circuit, and continue observing the temperature every half minute until it reaches its maximum value and begins to fall off regularly. Continue the observations for five minutes more to determine the rate of the fall of the temperature. The stirrer should be worked continuously but not too rapidly throughout the test, being careful, however, that the thermometer is not broken. When the observations have been finished, the conical-seated valve should be opened first to relieve the pressure and then the cover or cap can be unscrewed and removed.[1]

The method described for the use of the Mahler bomb calorimeter can be applied also for determinations of calorific value of liquid fuels. Heavy oils can be weighed directly in the platinum dish or crucible, but light oils which are easily vaporized must be put into specially prepared glass bulbs which are broken to allow access of the oxygen,

[1] Some engineers wash out the inside of the bomb with a little distilled water to collect the nitric and sulphuric acids formed. Usually, however, this correction for acids is not made, as the heat liberated in the formation of the acids is usually less than one-third of one per cent, which, of course, would be subtracted from the calorific value obtained. If the reader is interested he will find the method explained with the necessary data in the *Calorific Power of Fuels*, by H. Poole, page 62.

just before the cover is put on the bomb. If sufficient oxygen is provided in every case there will be complete combustion in the calorimeter with no other refuse than the cinders remaining.

A specimen calculation is given below:

Weights, — coal, .0030 lb.;[1] water in calorimeter, 4.85 lbs.; water equivalent of bomb, etc., 1.10 lbs. Weight of iron wire, .0002 lb.

Preliminary Observations.

Beginning	60.23° F.,	3 minutes, 60.26° F.,
1 minute,	60.24° F.,	4 minutes, 60.27° F.,
2 minutes,	60.25° F.,	5 minutes, 60.28° F.

$$\text{Rate of variation before combustion } a_0 = \frac{60.28 - 60.23}{5} = .01° \text{ F.}$$

Observations during Combustion.

6 minutes, 65.45° F.,
7 minutes, 67.29° F.,
8 minutes, 67.38° F., max.[2]

Observations after Maximum was reached.

9 minutes, 67.34° F.,	12 minutes, 67.28° F.,
10 minutes, 67.32° F.,	13 minutes, 67.27° F.
11 minutes, 67.30° F.,	

$$\text{Rate of variation after maximum } a_m = \frac{67.38 - 67.27}{5} = .022° \text{ F.}$$

The rate of variation of temperature before combustion was for cooling the water and that after combustion was for a loss of temperature by the water. Evidently, then, the two rates are opposed in effect and the true **average rate of variation** is

$$a_v = \frac{-.01 + .022}{2} = +.006° \text{ F. per minute.}$$

Three minutes (5–6, 6–7, and 7–8) were required for complete combustion or for the water to reach the maximum temperature. Total cooling correction to be added to the observed rise in temperature is, therefore, $3 \times .006 = .018°$ F.

[1] The coal had been warmed for one hour at a temperature of from 240 to 280 degrees Fahrenheit before weighing, in a crucible over a Bunsen burner or an alcohol lamp to drive off the moisture. In the best modern practice determinations are made on the basis of dry coal.

[2] Some engineers make a curve of temperatures (ordinates) and time (abscissas) and use for the final temperatures in the calculations the value from the curve, when the part of the curve representing the cooling becomes a straight line. The difference in numerical values by the two methods is usually very slight.

The total rise as corrected is $7.10 + .018 = 7.118°$ F.

The quantity of heat generated is, therefore, $Q = (4.85 + 1.10) \times 7.118 = 42.35$ (B.t.u.) for .0030 lb. of coal; and from this result

FIG. 258. — Atwater's Fuel Calorimeter.

FIG. 259. — Emerson's Fuel Calorimeter.

must be subtracted the heat of combustion of the iron wire $.0002 \times 3000$[1] or 0.60 B.t.u. The net value of the heat generated from the coal is, therefore, $42.35 - .60 = 41.75$ B.t.u.

A modification of the Mahler bomb calorimeter has been designed

[1] The calorific value of pure iron is about 3000 B.t.u. per pound, and iron wire No. 34 B. & S. gage one inch long will generate in combustion .63 B.t.u. This is the size of wire generally used in calorimeter work.

by Atwater,[1] **Fig. 258**, and another by Emerson, **Fig. 259**. The **former** consists of the shell of the bomb **A**, the cap **C** screwed on numerous threads to the shell, and holding down the cover **B**. Into the vertical neck of this cover a screw **E**, holding another screw **F**, is fitted and is to be turned down tightly, a lead washer serving as "packing." A small passage for the admission of oxygen from **G** is opened and closed as required by turning the screw **F** operating a needle-valve. A wire **H** of platinum or other non-oxidizable metal passes through the cover **B** and is insulated from it by a collar of hard rubber. Another wire rod **I** is attached to the lower side of the cover and electrical connection is made between the two wires **H** and **I** by a small iron wire stretched between them. A platinum crucible provided for receiving the fuel is supported by a "screw" ring. Ball bearings of hard steel are sometimes placed between the cover and the cap to reduce friction when screwing down. Holes located in the sides of the cap are for the attachment of a long spanner wrench when turning down the cap. A hand-stirring device **S** is used for agitating the water in the vessel **Q**.

FIG. 260. — Apparatus for Charging a Bomb Calorimeter with Oxygen.

The usual arrangement of the oxygen tank, pressure gage and tubing for charging a bomb calorimeter is illustrated in **Fig. 260**. A pellet press for compressing samples of fuel into a suitable **size** to burn in the crucible of this calorimeter is shown in **Fig. 261**.

Fig. 259 shows another form of bomb calorimeter (Emerson) of which the Mahler is typical. It consists of a nearly spherical shell **S**, divided into two parts which are screwed together by the ring **R**. Powdered fuel is placed in the crucible **C** and is ignited electrically by the current passing through the water in the vessel **Q** from the terminal at **A**, then through an insulated contact point **P** in the bottom of the calorimeter to a small platinum or iron wire in the crucible **C**, which becomes heated by the passage of the current to a white heat,

FIG. 261. — A Pellet Press for Compressing Samples of Fuel.

[1] Atwater, *Bulletin* No. 21, U. S. Dept. of Agriculture.

igniting the fuel. One end of this small wire is fastened to make electrical contact with the lining of the calorimeter, which in turn is connected electrically with the plug and terminal at **B.**

The outer vessel **O** is to be filled with water to the top. The stirring device consists of small propellers **F**, on a vertical shaft operated by a small electric motor **M.**

Fig. 262. — Typical Parr Calorimeter.

Fig. 263. — Parr Bomb for Hot Tube Ignition.

Fig. 264. — Parr Bomb for Electrical Ignition.

Parr Calorimeter. It is not always convenient to secure a supply of oxygen under pressure for use in a Mahler bomb, and consequently another type of fuel calorimeter, known as Parr's using a chemical (sodium peroxide) as the source of oxygen, has found considerable use, especially for relative determinations in power plants. The results obtained can scarcely be depended on to be as accurate as determina-

tions with one of the bomb type.[1] **Fig. 262** illustrates a simple form of Parr calorimeter. Sectional views of the two kinds of calorimeter vessels used are shown in **Figs. 263** and **264**. In the former the ignition is accomplished by dropping a hot wire through the neck into the shell A of the calorimeter. The cover is attached to the shell by means of a threaded nut F. A charge for the bomb consists of about .002 to .004 pound of pulverized coal from which the moisture has been driven off by warming for about an hour at a temperature of about 240 to 280 degrees

FIG. 265. — Parr Calorimeter with Motor Stirring Device.

Fahrenheit, and eighteen times as much by weight of sodium peroxide, which supplies the oxygen needed for combustion. Reactions produced by combustion are complex as the products of combustion CO_2, H_2O, SO_2, SO_3, etc., combine with some of the Na_2O_2 or Na_2O to form Na_2CO_3, $NaOH$, Na_2SO_4, etc. The charge should be well mixed

[1] As regards the accuracy of the determinations very much depends on the chemical purity and dryness of the sodium peroxide. If it is absolutely pure and the apparatus is handled with skill, it is easily possible to get results comparable with the bomb types.

by shaking the shell after the cover has been securely fastened. The cover must be attached very securely by turning up the nut with a long wrench while the shell is held in a vise or in some similar manner, because there is a violent explosion when ignition takes place. When the hot wire (**Fig. 263**) is put into the tube in the long neck **L**, the cap **R** at the top must be struck quickly with a mallet in order to open the valve **M**, which opens inward into the shell and permits the wire to fall through before it cools. To be certain of obtaining a good result the wire should be heated almost to a white heat, and the coal should preferably be put into the shell after the sodium peroxide. The rise of the mercury in the thermometer will indicate when an explosion has occurred.

The calorimeter is provided usually with wings or small propeller blades for agitating the water in the vessel **O**. The small pulley **P** (**Fig. 262**) shown at the top of the neck is used for turning the calorimeter bodily in the water when supported on the pivot **F**, shown at the bottom of the figure. The water **equivalent** of the calorimeter is determined in the same way as for other fuel calorimeters. **Fig. 265** shows the Parr calorimeter as designed for electrical ignition and with a stirring device operated by an electric motor.

Allowance must be made in the calculations for the heat generated by the chemical reactions of the sodium peroxide, which for the proportions given for a charge is approximately 27 per cent of the heat generated.

Example Illustrating Calorific Determination with Parr Calorimeter. Weight of powdered coal .0040 lb., containing 1.3 per cent moisture, and 6.7 per cent ash.

Weight of sodium peroxide	.072 lb.
Weight of water	7.160 lbs.
Water equivalent of bomb	0.253 lb.
Total equivalent weight of water	7.413 lbs.
Temperature rise	10.2°F.

Total heat generated,

$$7.413\times10.2 = 75.61 \text{ B.t.u.}$$

Heat due to combustion of Iron Wire (see page 215) 1.11 B.t.u.

Heat due to coal alone,

$$(75.61 - 1.11)(1.00 - .27) = 54.39 \text{ B.t.u.}$$

Heat value per lb. coal as fired,

$$\frac{54.39}{.0040} = 13{,}598 \text{ B.t.u.}$$

Heat value per lb. dry coal,

$$\frac{13{,}598}{.987} = 13{,}780 \text{ B.t.u.}$$

Heat value per lb. combustible,

$$\frac{13{,}780}{.933} = 14{,}780 \text{ B.t.u.}$$

Carpenter's Calorimeter. There are few calorimeters in which the oxygen is supplied at constant pressure which are altogether successful. One of the best forms of an apparatus of this kind has been designed by Carpenter, especially for coal determinations. With this apparatus no thermometers are needed, as the rise in temperature is measured by the expansion of the mass of water surrounding the combustion chamber.

This apparatus is shown in **Fig. 266.** It consists of a combustion chamber 15, provided with a removable bottom 17, through which the tube 23, supplying the oxygen, passes into the combustion chamber. Electric current for ignition is conducted through the wires 26 and 27. The removable bottom supports also the asbestos cup or crucible 22, used for holding the sample of coal to be burned. Just beneath the crucibles a silver mirror 38 is provided to "deflect" the heat. The plug containing the wires and the oxygen pipe 23 is made of alternate layers of asbestos and vulcanite. Products of combustion leave the combustion chamber through a spiral tube, the parts of which are marked 28, 29, 30, and 31, into the small vessel 39, attached to the outer casing of the instrument, and are finally discharged into the air from a small hole 41 in the side of the vessel. The pressure in the chamber 39 is indicated by a manometer gage 40. The inner casing of the instrument 1, containing the water for absorbing the heat generated, is nickel-plated and highly polished to reduce radiation as much as possible.

Fig. 266. — Carpenter's Calorimeter.

An open glass water gage 10 passes through the casings and extends below the water level. This water gage, with the scale attached to it, replaces the thermometer used in other calorimeters for measuring the rise in temperature. The scale is graduated to read inches, and it is calibrated usually by burning coke in the calorimeter, and determining thus the rise of the water level for a determinable weight of pure carbon. A calibration curve is usually supplied with the instrument. By mov-

ing the diaphragm 12, by means of the screw 14, the water level can be regulated, as well as the "zero" level in the glass water gage 10. A funnel 37 is provided for filling the instrument, and by inverting it, this funnel can be used also for draining. The instrument holds 5 pounds of water, and 2 grams of coal is the amount taken usually for a charge, requiring about twenty minutes for complete combustion of powdered coal. The asbestos cup 22 should be heated in the flame of a Bunsen burner before it is weighed. The charge of dried coal should then be put into it and weighed again. The difference will be the weight of the coal used. Now put the charge into the combustion chamber 15, place the platinum ignition wire above the coal, connect wires 26 and 27 to the battery, and as soon as the heat generated causes the level of the water to rise in the glass water gage 10, open the valve in the pipe discharging oxygen into tube 23, and then by pulling down the platinum wires to touch the contents of the crucible, the coal will be kindled. At the same time the reading of the glass scale opposite the gage glass 10 must be observed and recorded. Progress of the combustion can be observed through the glasses 33, 34, and 36, arranged vertically over each other for this purpose. As soon as the combustion is complete observe the time and the reading of the scale opposite the glass water gage 10. The difference between this last reading and the one taken at the beginning of the test is called the "actual" scale reading.

The Correction for Radiation is made by observing the reading of the scale of the water gage after the oxygen has been shut off, for a length of time equal to that required for the combustion. The difference between this reading and the "actual" reading is to be added to the "actual" reading to obtain the corrected reading.

By weighing the asbestos cup after the test is finished and subtracting from this weight that obtained previously for its weight empty, the weight of ash is determined.

In order that Carpenter's calorimeter may give determinations of heat values that are at all accurate, all the air must be removed from the water used, as the presence of air[1] will affect the relative level of the water in the gage glass for a given rise in temperature. The oxygen must also be supplied at a constant pressure, maintaining the pressure indicated by the manometer gage at the value for which the calorimeter was calibrated. Most calibration curves are made for a pressure of about 10 inches of water. The apparatus can be made to give good comparative results when operated carefully and "according to directions," but the general experience has been that calorimeters of this

[1] About two inches of kerosene oil are usually put into the glass water gage to prevent air from coming in contact with the water.

type give values that are from one to two per cent too low compared with results with a bomb calorimeter due to incomplete combustion. In general, the statement is often made that coal calorimeters intended for combustion at constant pressure will usually give nothing more than "faint approximations" to correct results.

When making calorific determinations of coal the distinction must be carefully made between results obtained per unit weight of combustible or per unit weight of coal (including moisture and ash).

Junkers' Calorimeter for Liquids and Gases. An apparatus for determining the calorific value of gases is shown in Fig. 267 and Fig. 268. The gas flowing in a pipe at the left (Fig. 267) passes through the meter **A**, then through the regulator **B**, and is burned in a type of Bunsen burner **C** in the lower part of the calorimeter. This instrument consists of a cylindrical copper vessel through which water is constantly circulating. The gases from the Bunsen flame in the calorimeter pass up through the hollow central portion of the instrument and near the top are deflected downward through a group of small tubes arranged in an annular ring between the outside and inside walls of the calorimeter.[1] Around these tubes water is kept circulating continuously to absorb the heat generated by burning the gas tested. After leaving these tubes the products of combustion discharge first into a chamber 31 (Fig. 268) and then into the air through the flue **D**. In order to keep the flow of water as regular as possible it is brought from the supply pipe **G** into a small reservoir in which the water is kept at a constant level (constant head) by means of an overflow pipe **H**. The water supplied to the calorimeter passes down through the pipe **6**, through a valve at **I**, and discharges at **K**, running into a vessel in which it is weighed. A graduated tube **Q** (Fig. 267) is provided to collect the

Fig. 267. — Junkers' Calorimeter with Auxiliary Apparatus.

[1] A modification of Junkers' calorimeter is made by the American Meter Company of New York and Philadelphia. In principle it is exactly the same as the one described, but is much improved in mechanical construction, the greatest advantage being that the nest of small tubes can be readily removed for repairs. In the original design removal for repairs is difficult if not almost impossible. The "American" type is used exclusively by the U. S. Bureau of Standards, which is a sufficient guarantee of reliability.

moisture from the steam that is condensed. The condensed steam collects in the combustion chamber 31 and escapes through the tube 35. A thermometer N, in a cup near the valve I, indicates the temperature of the water entering the calorimeter, and one at M shows the temperature of the water leaving. The temperature of the products of combustion (burned gases) is indicated by the thermometer O in the gas flue. The calorimeter is provided with an air jacket and is covered with sheets of copper, nickel plated and highly polished so that the radiation loss is considered negligible. If, then, the flow of water and the rate of burning the gas are regulated so that the **temperature of the products of combustion as indicated by the thermometer at O is the same as the temperature of the air surrounding the calorimeter,** practically all the heat generated by the burning gas is absorbed by the water. The rise in temperature of the water is observed by reading the thermometers at N and M.

Fig. 268. — Section of Junkers' Calorimeter.

Now if the temperatures of the water at the inlet and the discharge have been observed and the weight of the water flowing has been determined while, for example, a cubic foot of gas has been burned, then the difference in temperature in degrees Fahrenheit times the weight of water in pounds gives the heat value in British thermal units per cubic foot of gas. This is called the **higher heat value** of the gas.

Results of calorific determinations of a gas should be stated in a report as calculated as heat units per cubic foot of gas for the **standard conditions** of pressure and temperature. The American Society of Mechanical Engineers has favored the adoption[1] of **30 inches of mercury pressure** (14.7 lbs. per sq. in.) **and 62 degrees Fahrenheit,** while chemists and European engineers use as standard 29.92 inches (760 mm.) of mercury pressure and 32 degrees Fahrenheit (0 degrees Centigrade). This conversion can be readily made because the volume of the gas is directly proportional to the temperature and inversely to the pressure.

For some calculations relating to the efficiency of heat engines it is desirable to know the number of heat units representing the calorific

[1] *Journal A.S.M.E.*, Nov., 1912, pages 1795–1801.

value of the gas when the steam formed in the combustion is not condensed but is carried off with the products of combustion as is the case in practice. To determine this value, sometimes called the "**lower" heat value** of the gas, the latent heat at atmospheric pressure of the amount of condensed steam collected in the Junkers calorimeter must be subtracted from the value obtained by multiplying together the rise in temperature and the weight of water used. This correction is usually about five to ten per cent, having usually the smaller value for producer gases with high percentages of CO.

When thermal efficiencies of gas engines are calculated, it should always be clearly stated whether the "higher" or the "lower" heat value of the gas has been used. In all the codes of the American Society of Mechanical Engineers the "higher" heat value has the preference, but this is not by any means the generally accepted practice.

This apparatus, although it operates by a constant pressure method, gives very satisfactory determinations. Radiation loss is small and is neglected. Since tests with this apparatus are started when all parts are already heated normally, no water equivalent is to be taken into account. If the temperature of the discharge gases is not the same as that of the air supplied the results will be in error but the amount of this correction (see page 252, footnote) and the method of computing it is uncertain. For this reason the temperature of the discharge gases should be regulated most carefully and not more than two or three degrees Fahrenheit difference should be permitted between the temperature of these gases and that of the air supplied. Often it is necessary to open the windows of laboratory rooms to secure the proper temperatures. Temperatures of the water should be practically constant before a test is started.

As the Junkers calorimeter is ordinarily operated it does not determine accurately the "**higher**" heating value even if all the precautions stated have been observed. It is because an excess of moisture above that which came in with the air goes off in the discharged gases. The only way to eliminate this error is to supply gas and air that are saturated with moisture. The calorimeter will then give the true "higher" heating value, because all the moisture resulting from the combustion of hydrogen will be condensed, and will give up its latent heat. When a wet-gas meter is used it may be assumed that the gas is saturated as it comes to the apparatus. The obvious way to eliminate this error is to supply air which is also saturated. A convenient design is to connect the closed top of a cylindrical vessel about two feet high and five inches in diameter by a one-inch rubber tube to the bottom of the calorimeter, which, except for the opening for this tube, has been made air-tight. The cylinder is provided with a water-waste cock at the bottom. Several trays

covered with coke are placed inside the cylinder which is perforated with a number of half-inch holes around the perimeter near the bottom for the admission of air. A water-jet discharges from the top of the cylinder and the water trickles down over the coke as the air enters at the bottom and passes up to the air-pipe leading to the calorimeter. By this method air is thoroughly saturated and not only more accurate but also much more consistent results for the "**higher**" **heat values** are obtained. Obviously this humidity correction has no effect at least in the theory of combustion on the "**lower**" **heat values,** as they are calculated for the condition when all the water vapor due to combustion leaves in the discharged gases. This is one reason why many engineers prefer to base calculations of thermal efficiency on the "lower" heat values.

The error due to humidity can, however, be calculated approximately and the results correspondingly corrected. Moisture carried in the air can be determined by a wet- and dry-bulb thermometer (see page 368) and then assuming the discharged gases and the gas burned are saturated, the excess of condensation carried away in the discharged gases are readily calculated since their weight can be determined by a laborious calculation involving the computation of the weight of air supplied which must be obtained from the analysis of the products of combustion (discharged gases).

For making determinations of the calorific value of suction producer gas where the working gas pressure is less than atmospheric, a good method is to collect a sample with an aspirator and collecting bottle as explained for sampling flue gas (see page 236).

Producer gas and other gases of low heating value can be mixed with a little air and burned in a simple metal tube, covered over at the end with a piece of fine gauze to prevent firing back into the mixing chamber. The mixture formed should preferably be non-explosive.

The author has made continuous recording gas calorimeters using a simple pipe burner and positive pressure blowers (similar in design to those on page 366). One blower for measuring the gas is of about one-sixth the capacity of the larger one for measuring the air. Both blowers being driven by a single electric motor will always deliver gas and air in a constant ratio, provided pressure and temperatures of gas and air are maintained at about the values at which the instrument was calibrated. Calorific value of the gas is then proportional to the difference in temperature between the air and gas entering and the temperature of the discharged gases. A differential recording thermometer with the chart graduated in heat units per cubic foot of gas as determined by comparison with a Junkers calorimeter gives a continuous record of heat values of the gas. An apparatus of this kind is very useful in a

producer gas plant in showing the quality of gas produced and the relative care observed in the operation of the producers.

Exercise. Calorific Value of Gas. One cubic foot of coal gas at an absolute pressure of 28.9 inches of mercury and at 70 degrees Fahrenheit when burned in a Junkers calorimeter raised the temperature of 8.36 lbs. of water from 57.7 to 121.4 degrees Fahrenheit. Weight of condensation (water) collected due to combustion was .056 lb. Absolute atmospheric pressure 14.2 lbs. per sq. in. and corresponding latent heat of steam from tables = 971.5 B.t.u. per pound.

"Higher" Heat Value at "Room" Conditions, B.t.u. per cu. ft. =

$$(121.4 - 57.7) \times 8.36 = 532.5.$$

"Lower" Heat Value at "Room" Conditions, B.t.u. per cu. ft. =

$$532.5 - (971.5 \times .056) = 478.2.$$

Volume of Same Gas at Standard Conditions (30 in. mercury press. and 62 degrees Fahrenheit) =

$$\frac{1 \times 28.9 \times 522}{30 \times 530} = 0.949 \text{ cu.ft.}$$

"Higher" Heat Value at Standard Conditions, B.t.u., per cu. ft. =

$$\frac{532.5}{.949} = 562.$$

"Lower" Heat Value at Standard Conditions, B.t.u. per cu. ft. =

$$\frac{478.2}{.949} = 504.$$

Fig. 269 shows a balance and lamp attachments for a Junkers calorimeter set up for determining the heat value of liquid fuels like gasoline, kerosene, crude oil, etc. The heat generated is measured in the same way as when gas is burned and the weight of oil used is determined by weighing on the balance to which the lamp **L** is attached. This lamp is provided with a "regenerative" burner **B** with a long stem as shown. A small hand pump **P** is arranged for attachment to the valve **V** to put the oil in the bowl of the lamp under a pressure of about ten pounds per square inch. This air pressure forces the oil up the stem and through a coil of metal tubing which lies above the flame and is heated by it and gasified. The gaseous fuel escapes as a jet through a minute orifice where it should burn with a

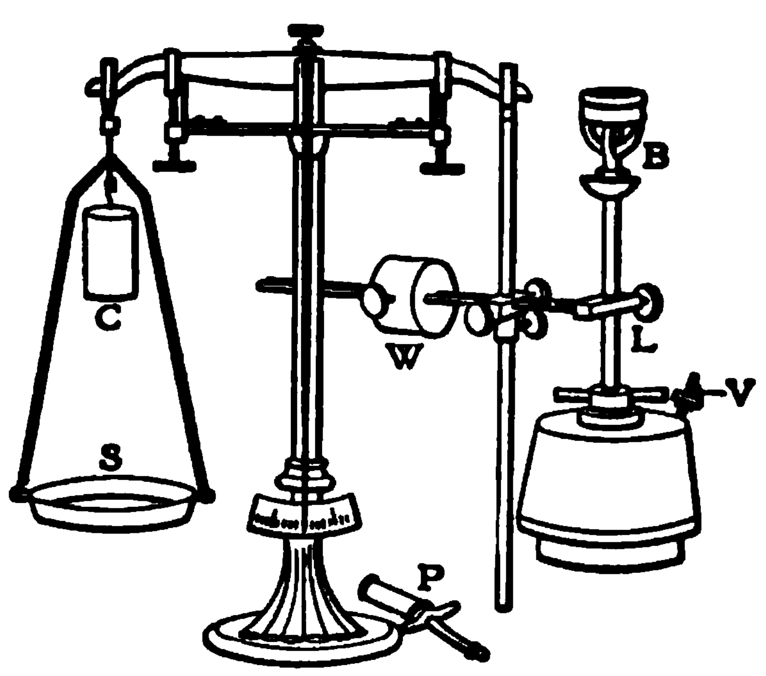


Fig. 269. — Balance and Lamp for Burning Oils in Junkers Calorimeter.

blue flame indicating perfect combustion. When using oils heavier than gasoline the regenerator coil must be heated by burning alcohol in the cup shown in the figure just below the burner **B.** When combustion is not complete as is always the case when the flame is started soot accumulates on the coil and is likely to choke the orifice of the burner. A piece of fine piano wire should always be at hand for cleaning the orifice.

Calorific Values from Chemical Analysis. Dulong stated a long time ago that the heat generated by burning any fuel was equal to the sum of the " possible heats " generated by its component elements, less that portion of the hydrogen which combined with the oxygen in the fuel to form water. When hydrogen and oxygen exist together in a compound in the proper proportions to form water, the combination of these elements has no effect on the calorific value of the compound. Now the calorific value of a pound of carbon is 14,600 B.t.u. and of a pound of hydrogen is 62,000 B.t.u., so that by Dulong's formula, the calorific value of a pound of fuel **x** would be stated, using these values, as

$$\mathbf{x} = 14{,}600\,\mathrm{C} + 62{,}000\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right) + 4{,}000\,\mathrm{S}, \quad . \quad . \quad . \quad (69)$$

where **C, H, O** and **S** are respectively the weights of the carbon, hydrogen, oxygen and sulphur in a pound of fuel.

A similar formula known as that of the *Verein deutscher Ingenieure* expressed in units and terms used in Dulong's, but corrected for **w** per cent of moisture is given as follows:

$$\mathbf{x} = 14{,}400\,\mathrm{C} + 62{,}000\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right) + 4{,}500\,\mathrm{S} - 1{,}100\,\mathbf{w}. \quad . \quad (70)$$

Formulas given above are all for the " higher " heat values corresponding to those obtained with a bomb calorimeter. The last formula (70) expressed for " lower " heat value is:

$$\mathbf{x} = 14{,}400\,\mathrm{C} + 52{,}000\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right) - 1{,}100\,\mathbf{w}. \quad . \quad . \quad . \quad (71)$$

As the result of testing forty-four different kinds of coal with his bomb calorimeter Mahler developed the following formula, using the same symbols used in Dulong's,

$$\mathbf{x} = 200.5\,\mathrm{C} + 675\,\mathrm{H} - 5{,}400. \quad . \quad . \quad . \quad . \quad . \quad (72)$$

Using this latter formula Lord and Haas[1] computed the calorific values for a series of 40 Pennsylvania and Ohio coals which they had analyzed and found that the maximum differences between the calculated results and the determinations with a bomb calorimeter were from 2.0 to − 1.8

[1] *Trans. American Inst. of Mining Engineers*, Feb., 1897.

per cent. With fuels like coke, charcoal, and anthracite coal, in which the content of volatile matter is small, the calorific values calculated from an accurate analysis are usually in very close agreement with accurate calorimeter tests, but with coals having more than 20 per cent of volatile matter there is likely to be considerable error.

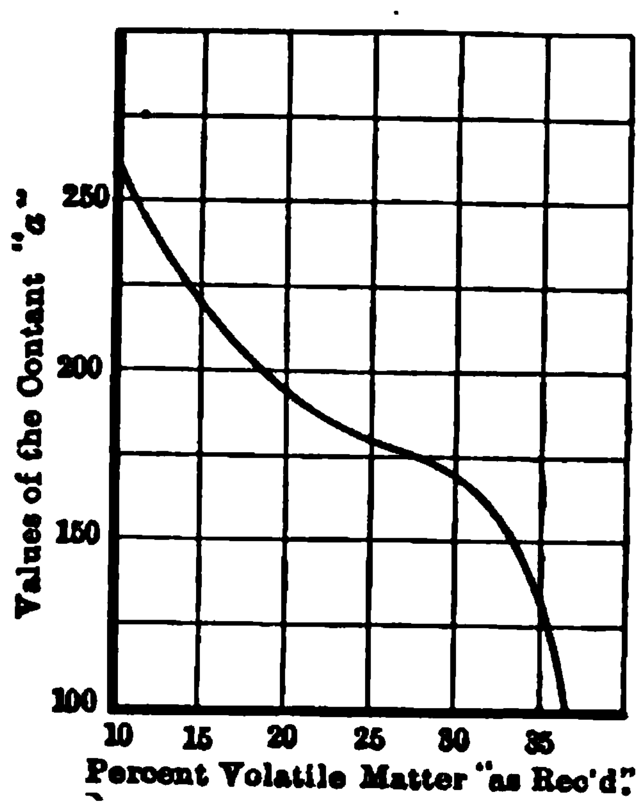


FIG. 270. — Curve for Determining Calorific Value of Coal.

A formula which is as accurate as any of those given above is based on the results of the proximate analysis. In this formula, known as Goutal's,[1]

c = per cent fixed carbon in coal "as received."

v = per cent volatile matter in coal "as received."

a = constant from the curve in **Fig.** 270, then,

$$x = 147.6\,c + w. \qquad (73)$$

Proximate Analysis of Coal. For all tests in which an analysis of the coal or its calorific value is to be determined, it is very necessary that the sample to be tested be selected with the greatest care. The method generally adopted for obtaining a fair sample is known as "quartering," as explained in the **Rules for Conducting Boiler Tests** adopted by the American Society of Mechanical Engineers. (See page 231.) The utmost care must be taken that the amount of moisture in the sample received for analysis is the same as that in the original condition, or more specifically in a **boiler test,** at the time when the coal used in the test was weighed. For this reason samples of coal should be transported and stored in air-tight preserving jars or similar vessels. It is not unusual, moreover, to find that coal containing 10 per cent of moisture will lose as much as 2 or 3 per cent of its moisture in the process of careless sampling, crushing, resampling, etc., while if it is allowable to remain exposed to atmospheric conditions for a considerable time in a warm room as much more may be lost by evaporation. Crushing, sampling, and weighing should always be done as rapidly as possible.

Only during recent years has the proper importance of correct sampling of coal for analysis been understood. Particularly in run-of-mine coal, that is, coal as mined without crushing or screening, the careful selection of coal for the sample as regards size is also very important. If the sampling has not been done so as to get coal that is representative, certainly the analysis can be of no value. Proportionate amounts should be

[1] *Wisconsin Engineer,* Dec., 1911.

taken of both large and small sizes as well as of the fine dust. In the best engineering practice to-day at least 200 pounds of coal is collected for the process of sampling for the analysis. This amount of coal is to be broken up on a clean floor by any convenient means to a size of about ½ inch diameter, then thoroughly mixed and spread out on a flat circular pile. This pile is then "quartered," and opposite quarters are discarded. The remainder is now further broken up to about ¼ inch diameter and the mixing, quartering and discarding is continued until from five to ten pounds remains. This is to be put into a glass jar or a tin can that can be made air-tight. The sealing should be carefully done, to prevent any deterioration of the sample in transportation to the laboratory where the analysis is to be made.

In the laboratory the coal should be emptied from the jar or can and crushed to a fineness of about a 20-mesh sieve (20 meshes to the inch). The crushed coal is thoroughly mixed and a small portion, about 2 or 3 ounces, is put into an air-tight bottle and is to be used for the analysis. The rest is put back into the jar or can and sealed. It is to be retained for possible use in check tests.

Proximate analysis of coal consists in determining the moisture, volatile matter, fixed carbon, ash and sulphur. Methods for these determinations are more or less empirical and vary slightly, so that in a report the authorship of the methods used should be stated. The methods most generally accepted by progressive engineers are those defined by the American Chemical Society,[1] and are in common use both in this country and in England.[2]

Moisture and ash determinations are most important because they are non-combustible are detrimental constituents, the moisture requiring the wasting of heat for its evaporation into steam and the ash when present in large amounts is often likely to form clinker in poorly designed or badly operated furnaces and is also expensive to dispose of. Sulphur determinations are important only when the furnaces are not suitable for the combustion of coal containing one or two per cent of sulphur. Furnaces are readily designed for burning coals having from four to five per cent of sulphur without clinkering the ash.

Methods of the American Chemical Society are as follows : —

1. **Moisture.** Weigh in a covered crucible about four grams (about ⅛ oz.) of the coal passing through a 20-mesh sieve that was prepared for analysis as described above. This should be done as quickly as possible to avoid loss of moisture to the air. Remove the cover and heat in an oven for an hour at a temperature of from 220 to 230 degrees Fahrenheit. At the end of the hour replace the cover on the crucible, remove it from

[1] *Journal of American Chemical Society*, vol. 21 (1899).

[2] *The Testing of Motive Power Engines*, by R. Royds (London, 1911), page 293.

the oven and place it in a desiccator to cool.[1] When the crucible and the coal it contains are at nearly the same temperature they should be weighed. Again remove the cover and heat as before in the oven for a half hour longer. If the weight has remained constant no more heating is necessary and the difference between the first and last weighings is the moisture in the coal.

2. **Volatile Matter.** Weigh about one gram ($\frac{1}{30}$ oz.) of the "20-mesh" crushed coal prepared for analysis in a platinum crucible weighing from 20 to 30 grams ($\frac{2}{3}$ to 1 oz.)[2] having a closely fitting cover. Support the covered crucible on a chemist's triangle of nichrome steel or of platinum which should be about 3 to $3\frac{1}{2}$ inches above the top of a good Bunsen burner.

Heat the covered crucible for seven minutes in the full flame of the burner which should be at least eight inches high when unobstructed. Cool the crucible in a desiccator and then weigh carefully. The loss is the sum of the volatile matter plus the moisture. The room in which the test for volatile matter is made should be free from drafts which might cause a variation in the intensity of the flame.[3]

3. **Fixed Carbon and Ash.** The crucible without the cover and the residue from the preceding test are now intensely heated with a Bunsen flame or with an air-blast lamp until all the carbon has been burned, and the weight of crucible and contents becomes constant. The use of the air-blast lamp in place of the Bunsen burner for this last determination will very much reduce the time required. The contents of the crucible may be stirred slightly with a platinum wire to break up the ash which should

[1] While the sample is cooling there is the possibility that it may absorb moisture from the air unless it is placed in a desiccator until cool. It is difficult to get accurately the weight of hot bodies on account of the air currents produced.

[2] The capacity of the crucible should be about three times the volume of the coal to allow for its expansion in coking.

[3] It is not at all unusual for persons making tests on the same sample of coal to disagree as to the volatile content. This is because not all chemists and engineers will use the same type of burner, and if they are in different places they may have gas of widely different heat value. To overcome these difficulties the author has made electric furnaces consisting of a single resistance coil of nichrome wire calibrated for an impressed wattage to give a temperature of from 850 to 900 degrees Fahrenheit in a platinum crucible placed in its core. In five minutes after the electric current is applied to the coil a constant and maximum temperature is reached and the covered platinum crucible is then inserted to be heated for seven minutes. Weighings are made as with the regular method. Results show remarkably close agreement, and are independent of variable gas supplies.

Porcelain crucibles should never be substituted for platinum crucibles for accurate work as the porcelain requires a longer time to attain a constant temperature and therefore the duration of the application of the maximum temperature may not be the same as with one of platinum.

become a powdery mass when combustion is complete. Combustion is assisted by inclining the crucible on the triangle during this test so as to admit air more freely for oxidation. After cooling in a desiccator make a final weighing. The difference between this weight and that of the crucible without cover when empty is the ash. Weight of fixed carbon is obtained by subtracting the sum of the weights of moisture, volatile matter and ash from the original weight of the sample of coal tested.

Weighings should be made with chemical scales sensitive to $\frac{1}{1000}$ of the amount weighed. Two determinations of the complete proximate analysis should be made of each sample and the results should check within half of one per cent of the weight of the coal.

In a report record with the regular data whether in the volatile test the coal coked into a single spongy mass or whether it remained granular.

A. S. M. E. Methods. Methods proposed for poximate analysis of coal by the Power Test Committee of the American Society of Mechanical Engineers vary somewhat from the above. The following important items should be cited:

1. In tests where firing is done by hand select a representative shovelful from each barrow-load as it is drawn from the pile and store the samples in a cool place in a tightly covered metal receptacle.

When all the coal has thus been sampled, break up the lumps, thoroughly mix the whole quantity, and finally reduce it by the process of repeated crushing, quartering and discarding opposite quarters to a sample weighing about 5 pounds, the largest pieces being about the size of a pea. From this sample two one-quart air-tight glass fruit jars, or other air-tight vessels, are to be promptly filled and preserved for subsequent determinations of moisture, calorific value and chemical composition. These operations should be conducted where the air is cool and free from drafts.

When in the process of quartering and discarding the sample lot of coal has been reduced to about 100 pounds, a portion weighing say from 15 to 20 pounds should be withdrawn for the purpose of immediate moisture determination. This is placed in a shallow iron pan and dried on the hot iron boiler flue for at least 12 hours, being weighed before and after drying on scales reading to quarter ounces.

The moisture thus determined is approximately reliable for anthracite and semi-bituminous coals, but not for coals containing much inherent moisture. For such coals, and for all absolutely reliable determinations, the method to be pursued is as follows:

Take one of the samples contained in the glass jars and subject it to a thorough air-drying, by spreading it in a thin layer and exposing it for several hours to the atmosphere of a warm room, weighing it before and

after, thereby determining the quantity of surface moisture it contains. Then crush the whole of it by running it through an ordinary coffee mill or other suitable crusher adjusted so as to produce somewhat coarse grains (less than $\frac{1}{16}$ inch), thoroughly mix the crushed sample, select from it a portion of from 10 to 50 grams,[1] weigh it in a balance which will easily show a variation as small as 1 part in 1000, and dry it for one hour in an air or sand bath at a temperature between 240 and 280 degrees Fahrenheit. Weigh it and record the loss, then heat and weigh again until the minimum weight has been reached. The difference between the original and the minimum weight is the moisture in the air-dried coal. The sum of the moisture thus found and that of the surface moisture is the total moisture.

To determine volatile matter place one gram of air-dried powdered coal in the crucible and cover it with a loose platinum plate. Heat 3½ minutes in the flame[2] of the Bunsen burner, and continue the heating for 3½ minutes longer in the flame of the blast lamp. Cool down,[3] remove the cover, and weigh the residue. The loss in weight represents the combined volatile matter and moisture. Subtracting the moisture, the weight of volatile matter alone is determined.

To ascertain the ash expose the residue in the crucible to the blast lamp until it is completely burned, using a stream of oxygen if desired to hasten the process. The residue left is the ash.

The difference between the residue left after the expulsion of the volatile matter and the ash is the fixed carbon.

To determine sulphur by Eschka's method, which is the one commonly used, heat 1 gram of coal mixed with 1 gram of magnesium oxide and ½ gram of sodium carbonate for 1 hour, using an alcohol lamp. After cooling mix with 1 gram of ammonium nitrate and heat the mixture 10 minutes; then dissolve in 200 cc. of water, heat and reduce by evaporation to 150 cc., acidify with hydrochloric acid and filter. Add barium chloride to the filtrate and determine the sulphur by calculation from the quantity and composition of the barium thereby precipitated.

The carbon and hydrogen are obtained by the use of the combustion apparatus. One-half gram of the pulverized air-dried coal is placed in a porcelain "boat," which is introduced between the copper roll of oxidized copper gauze and the copper oxide within the glass combustion tube. After the coal and the entire contents within have been thoroughly dried out by a sufficient preliminary heating, aided by a current of dry air, the furnace is set to work and the coal burned by first passing

[1] About ⅓ ounce to 2 ounces.

[2] Height of flame is not specified. It could be stated that it should be in the hottest part of flame as in some coal specifications.

[3] Cool in a desiccator.

air through the tube and finally oxygen. The products of combustion are to be passed through potash bulbs and a chloride of calcium tube. The carbon dioxide produced by the combustion of the carbon is absorbed by the potash, and the water formed by the combustion of hydrogen, together with that due to the moisture in the air-dried coal, is taken up by the chloride of calcium. The quantity of carbon is determined by weighing the bulbs before and after, thereby obtaining the weight of the carbon dioxide produced, and then calculating the weight of carbon from the known composition of the dioxide. Likewise, the quantity of hydrogen is determined by weighing the calcium tube before and after, which, after deducting the moisture in the air-dried coal, gives the amount of water produced, and, dividing by 9, the amount of hydrogen.

The ultimate analysis of coal for sulphur, pure carbon and hydrogen, as will be seen from the above description, requires the use of so much chemical apparatus, and at best it is so complicated, that it is not likely to be done except in a fully equipped chemical laboratory. *It should not be undertaken by one who is not entirely familiar with all the details of the work.*

Hydrogen from Proximate Analysis. Professor L. S. Marks[1] has developed an empirical formula for the determination of the hydrogen in coal from the proximate analysis. In this formula **V** is the per cent by weight of volatile matter in the combustible part; that is, coal less the sum of moisture and ash, and **H** is similarly the per cent by weight of hydrogen in combustible, then,

$$\mathbf{H} = \mathbf{V}\left(\frac{7.35}{\mathbf{V} + 10} - .013\right).$$

Purchasing Coal by Calorific Value ("B. t. u's") and Analysis. Every year there are more power plants purchasing their coal on the basis of analysis and calorific value. It may be generally assumed that shipments of coal coming from the same mine at different times will not vary a great deal in the composition of combustible. Moisture and ash may however vary considerably and every large shipment, preferably every car-load should be tested at least for moisture and ash content as a check to determine whether the specifications are being fulfilled. At frequent intervals particularly when ash or moisture determinations indicate a doubtful quality, samples should be sent to some laboratory for a complete analysis. For only the moisture and ash determinations no special equipment is needed. If moderately large weights are used, a fairly accurate "counter" or even a good platform scales may be used for weighing; and aside from this only a few sheet metal or tin pans, one or two thermometers, and several Bunsen

[1] *Power*, Dec., 1908.

burners and porcelain crucibles are absolutely required.[1] Besides the scales for weighing the equipment needed can be purchased for ten dollars. For a drying box almost any kind of a discarded galvanized iron or tin vessel can be used. There should be a small hole in the top for the insertion of a thermometer.

To burn coal to ash with Bunsen burners requires the application of heat for a great many hours. Instead of providing air pressure for a blast lamp some engineers prefer to buy oxygen in tanks[2] and burn the coal rapidly in a pure oxygen atmosphere.

[1] For a more complete discussion of the sampling and analysis of coal as well as criticisms of sample specifications, see *Bulletin* No. 5, The Pennsylvania Engineering Experiment Station, State College, Pa. (Free distribution.)

[2] Linde Air Products Co., Buffalo, N. Y., with distributing stations in several large cities.

CHAPTER IX

FLUE GAS ANALYSIS

Flue Gas Analysis. The analysis of flue gases in connection with tests of steam boilers gives a valuable means for determining the relative value of different methods of firing and of different types of furnaces. Errors in the analysis of flue gases are most often due to the inability to secure an average sample of the gases in the different parts of a flue or chimney. The composition is likely to vary considerably even during short intervals, and it is therefore desirable to adopt some method of sampling which will permit collecting the sample slowly and continuously for a considerable period.

A very simple and convenient sampling apparatus is shown in **Fig. 275.** The sample of gas[1] is taken from the flue or chimney through the pipe **P** shown at the top of the figure. This pipe extends well into the flue and has usually a long slot cut into its side so that presumably a better sample of gas can be taken than if it were taken at the end of the pipe. The pipe outside the flue is connected by means of a short rubber tube to the sampling bottle. A valve **V** should be put as near as possible to the end of such pipes, so that they can be closed up when the sampling bottle is removed for analysis. If the pipe cannot be closed, the suction in the flue will draw air into the pipe and fill it so that when again connecting up the sampling bottle all this air must be removed before a true sample can be taken. The sampling bottle is preferably one with a wide neck, closed with a rubber stopper through which two glass tubes pass into the bottle, one reaching nearly to the bottom and the other entering only a little below the stopper. Tube **B** can be connected to an **aspirator** or **ejector**, or any similar device producing a suction, and there will be a steady flow of gas through the bottle. If still another short tube like **B** is put into the stopper it can be used for the attachment of gas analysis apparatus, making a very satisfactory arrangement, and it is then unnecessary to disconnect the sampling bottle from the pipe.

FIG. 275.—Gas Sampling Bottle.

Small aspirators or ejectors (**Fig. 276**) operating on the principle of

[1] For further discussion of sampling of the gas see "Boiler Code — Location of Instruments," page 272.

an injector with a small stream of water which entrains the gases is very convenient for collecting samples continuously. A slightly different design made of pipe fittings is shown in **Fig. 277.** Water enters through a vertical nozzle **N** and in discharging as a jet at high velocity entrains air or gas drawn in through the side opening and the mixture of atomized water and air is discharged with considerable velocity through the forcing tube **F** at the bottom.

If an aspirator is not available, the sampling bottle and the tube **B** may be filled with mercury. Then by gradually siphoning the mercury from the bottle the flue gases will be drawn in.

By adjusting the valve **V** the rate of flow of the gases into the bottle can be regulated. Mercury is too heavy to use in a very large sampling bottle so that water is often used instead, with the disadvantage, however, that the water will probably absorb some of the constituents of the gas. On this account very little water should be left in the bottle with the sample of gas. If the water is saturated with gases, as it will be from long use, this precaution is not so essential. Sampling and collecting bottles should have rubber rather then cork stoppers, because cork is too porous.[1]

Fig. 276. — Water-jet Aspirator or Ejector.

The sampling bottle and the tubes must be completely filled with the liquid before beginning to take the sample by the method of displacing water, because any air left in them will remain in the bottle and will be mixed with the sample of the gas. If the end of the stopper going into the bottle is made slightly conical it will be easier to avoid entrapping bubbles of air at the top of the bottle when inserting the stopper.

This type of sampling bottle when filled with liquid can be used also very conveniently by reversing the connections of its tubes; that is, by attaching the long tube to the pipe entering the flue, and then turning the bottle upside down. The liquid will then run out through the shorter tube and the gas will be drawn in to fill the bottle.

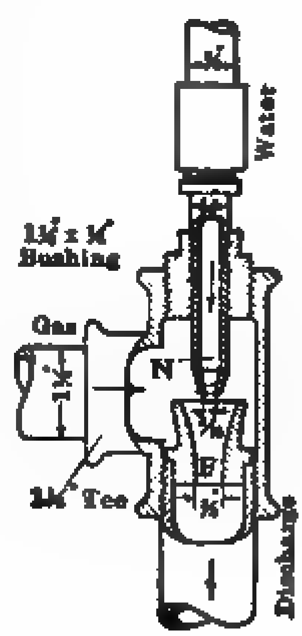


Fig. 277. — Ejector made of Pipe Fittings.

[1] A good way to cut holes in rubber stoppers for tubes is with the ordinary drills used for metal work, using a drill considerably larger than the hole required.

A portion of the gas can be removed from the sampling bottle into the measuring burette or tube required for making the analysis of the flue gas by connecting the short tube of the sampling bottle to the burette or to some other part of the gas analysis apparatus as may be required. If now the rubber tube connected to the longer tube of the sampling bottle when disconnected is put into a pail well filled with water, then as the gas is withdrawn from the bottle, water will be drawn from the pail to displace it. One of the advantages of this sampling apparatus is that it can be easily made from the materials obtainable in almost any town or village. A two-quart preserving jar with a rubber stopper to fit and tubes of glass, brass or iron can be used to make up a very good apparatus.

Fig. 278 shows a sampling device used extensively in England. It consists of an iron tube T of $\frac{3}{8}$-inch-iron pipe open only at the end in the flue. The end is located carefully however in English practice so that it lies well into the current of the gases. The pipe **T** is connected to a vessel **M** about two feet in diameter. The bottom is connected to another vessel of about the same size which is open at the top to the atmosphere. Samples for analysis are taken from the small bottle **A**. Vessels **M** and **N** are provided to prevent stagnation of the gases in the pipes. Rubber tubing is used in only very short lengths because it is said to be more or less porous to CO_2. When the test is started the bottle **A** is full of mercury, cock **C** is closed, and **D** is open just enough to permit a slow flow of mercury into **B**. When a new test is to be started the mercury in **B** is poured into **C** and **A** is filled and slowly emptied as before.

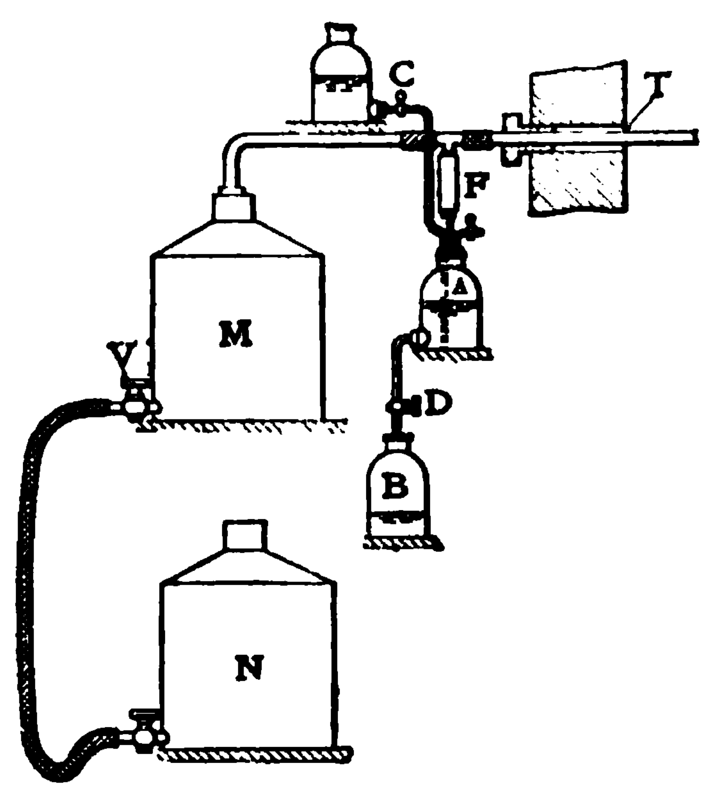


Fig. 278. — English Gas Sampling Apparatus.

Since there is nearly always a great variation in the composition of the gases in the various parts of a flue or chimney it is not very likely that a tube open at the end and having a long slit like the one described in the preceding paragraphs will give a "fair" sample. Obviously most of the gas will enter the slot in that portion of its length nearest the collecting apparatus. Another device often used for a sampling tube consists of a horizontal pipe into which a number of branch tubes are fitted. These branch tubes are arranged so that the openings at their ends will take samples from the different parts of the flue or chimney in which they are placed. Sometimes these branch pipes are also slotted or are perforated with small holes drilled into their walls.

Fig. 279 shows an arrangement of sampling tubes for collecting flue

gas recommended by the American Society of Mechanical Engineers in the code of 1899. It consists of a series of standard one-fourth inch pipes, all open and otherwise alike at the ends and of equal lengths. Each pipe is to be placed with one end in a shallow air-tight box or receiver made of sheet iron. It is convenient usually to make the depth of this sheet-iron box about the same as that of a course of bricks. These tubes should be arranged so that the open ends will be at points well distributed over the area of the flue which in the figure is marked **A.** The other ends, which are also open, are to be enclosed in the receiver **B.** The receiver is connected by four tubes **C, C** with a mixing box **D.** The flue gases drawn from it should be well mixed and should represent an average sample from the flue or chimney from which they are taken. Tests have shown that two such sampling devices placed in the same flue one above the other about a foot apart, will furnish samples of the flue gases showing the same composition when analyzed. There are several reasons why this apparatus having for a number of years the approval of the national societies was scarcely ever used. First, it is very expensive costing about $25 for each installation, and second, it is difficult to keep air-tight in large sizes. The recent recommendation of the Power Test Committee is that a single tube be used having perforations "extending the whole length of the part "immersed", and pointing toward the current of gas, the collective area of the perforations being less than the area of the pipe." Many American engineers of repute prefer a single pipe open only at the end like the one in **Fig. 278,** and observe extreme care in its location to get an average sample.

A very convenient type of sampling bottle is shown in **Fig. 280.** It consists of a bottle with an opening at the bottom (tubulated), and is provided with a stopper at the mouth through which a glass funnel **F** and a tube are passed. The bottle contains water and light oil, and when it is filled there will be a layer of about 4 inches of the oil over the water. The tube **O** at the top is to be connected to the sampling tubes in the flue and the sample is taken in by opening the valve in this tube and also the one at the bottom of the bottle. The water drains off at the bottom and is replaced by the sample of gas. The glass funnel is used for pouring water into the bottle and in this way expelling the gas needed for analysis. The gas is thus made to pass out through the same tube through which it is drawn in.

Another type of sampling apparatus used by many engineers consists of two galvanized-iron tanks, each about 2 feet high, and about 5 inches in diameter. On the side of each of these tanks and close to the bottom a valve is attached by soldering. These two valves are connected by a piece of heavy rubber tubing. One of the tanks is closed at the top, and a small stopcock or valve is attached to the cover. The other tank is

open at the top. The apparatus is used for collecting gas by filling to "overflowing" the tank with the closed top with water from the other tank by raising the latter so that the level of the water in it is above that of the water in the closed tank. By means of rubber tubing the stopcock or valve on the closed tank is then connected to the sampling tubes in the chimney or flues. Meanwhile the open tank is held at such an elevation that the water will not run back into it and create a vacuum in the closed tank. After this connection has been made the stopcock and valves are to be opened again, so that when the open tank is placed below the level of the closed one, the water will flow into the open tank

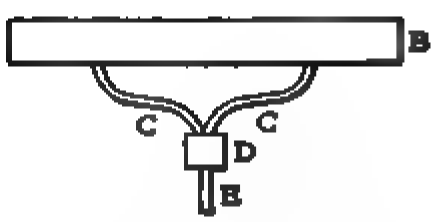


FIG. 279. — A.S.M.E. Arrangement of Sampling Tubes for Flue Gas.

FIG. 280. — Another Type fo Sampling Bottle.

and fill the other one with gas. This operation should be repeated several times before the sample is carried away to be analyzed, so that there can be no doubt that none of the air in the sampling tubes entered the sample to be analyzed. This water can be used over and over again, and when it has become saturated with gas it is practically as good as mercury for use in collecting the gases.

When a sample of flue gas is taken from the flue at a considerable distance from the furnace it is likely to become mixed with air leaking through the brickwork of the boiler setting, and the analysis will not show the true relations between the volumes of the so-called flue gases and the excess of air. To prevent as much as possible this leakage of air the joints in the masonry must be examined and repaired if necessary and the sample must be taken as near as possible to the fire, bear-

ing in mind, however, that they must be drawn very slowly from the hot flue in order that they will be cooled down gradually to avoid dissociation. For hot gases an earthenware collecting vessel may be used if a glass bottle is likely to be broken. If dissociation occurs in the sample the analysis may show results entirely different from the true composition of the gas in the flue. It is also difficult to prevent the entrance of air into a flue through the bearings of dampers, and whenever it is possible the sample of flue gas should be obtained between the furnace and the damper. At high temperatures sampling tubes of other metals than platinum[1] or nickel are not quite satisfactory, since by their oxidation they abstract the oxygen from the gases passing through them.

Apparatus for the Analysis of Flue Gases. Samples of flue gases contain in varying amounts carbon dioxide (carbonic acid), oxygen, carbon monoxide, nitrogen, unburned hydrocarbons, and occasionally some free hydrogen. For the data which an engineer usually requires it is not necessary to determine by direct analysis more than three of these; carbon dioxide, CO_2, oxygen, O_2, and carbon monoxide, CO.

The determination of CO with the facilities and the portable apparatus ordinarily available in engineering laboratories is often somewhat doubtful. Some authorities state that there is rarely more than a trace of CO to be found in the gases from combustion in the ordinary types of furnaces. When more than one per cent of CO is shown by the analysis and the CO_2 determination is not over 14 per cent, it may usually be assumed that a large part of what is taken to be CO is oxygen which was not absorbed by the proper reagent.

In the following table a set of analyses of flue gases is shown. The determinations were made by Scheurer-Kestner with coal from Ronchamp. Other analyses of flue gases may be checked by a comparison with this table. Thus when the analysis shows about 8.2 per cent CO_2, the sum of the percentages of CO_2 and O_2 will probably be between 19 and 20.

PERCENTAGE COMPOSITION OF FLUE GAS

CO_2	O_2	CO	N	Hydrocarbons.
8.2	11.3	.2	79.8	.5
10.8	9.0	.2	79.7	.3
12.9	5.5	.2	80.3	1.1
13.4	4.4	.2	80.2	1.8
14.6	2.8	.3	80.6	1.7

[1] Porcelain and annealed glass are also satisfactory materials to use for making sampling tubes for very hot flues.

In the portable apparatus used by engineers for the analysis of flue gases a separate pipette or treating tube is provided for each reagent, and the chemicals used are of greater strength than the reagents used by some chemists. The following reagents give satisfactory results in a portable apparatus:

(1) For absorbing CO_2 a solution of one part of potassic hydrate or caustic potash (KOH) dissolved in two parts **by weight** of water is generally used.

(2) For absorbing O_2 either an alkaline solution of pyrogallic acid[1] or sticks of phosphorus are employed.

The alkaline solution of pyrogallic acid is prepared by mixing together preferably in the absorption pipette or treating tube, to prevent access of air, 5 grams of pyrogallic acid powder and 100 cubic centimeters of potassic hydrate (KOH) solution prepared as explained above.

To make the absorption more rapid some engineers use a solution very much stronger in pyrogallic. This is not good practice as stronger solutions are likely to evolve CO in the presence of oxygen.

Phosphorus is more rapid in its action than the pyrogallate, but has the disadvantage of being difficult to use as it must be handled under water.

(3) For absorbing carbon monoxide a hydrochloric acid solution of cuprous chloride is used. This is prepared by dissolving about 10 grams of cupric oxide in from 100 to 200 cubic centimeters of concentrated hydrochloric acid. This solution must be allowed to remain in a bottle tightly closed and well filled with copper wire or gauze, until the cupric chloride is reduced to cuprous chloride. In this latter state the liquid will be colorless. Exposure to the air produces a brown color, indicating the cupric state.

After a time these reagents must be replaced by new solutions. The potassic hydrate solution may be used until each volume has absorbed forty volumes of CO_2. Pyrogallic acid solution deteriorates rapidly and each volume should be expected to absorb only one or two volumes of O_2. Cuprous chloride will absorb an equal volume of CO.

Portable devices for the analysis of flue gases are generally known as "Orsat" apparatus. Of these there are various types. The one devised by **Fisher**, shown in **Fig. 281**, has been used extensively. It consists of a measuring-tube **M** surrounded by a water-jacket, and set of absorption pipettes, **A, B, C,** each filled with a reagent. Each of these pipettes (**Fig. 282**) consists of two glass vessels connected by a U-shaped glass tube at the bottom. One end of these pipettes is joined by means of a short piece of rubber tubing to a glass yoke **T,** which is designed for

[1] When the temperature is lower than about 55 degrees Fahrenheit this reagent does not give satisfactory results.

attachment at one end to a tube leading to the sampling-bottle and at the other end to the measuring-tube. A water bottle **W** is connected by a flexible rubber tube **P** to the bottom of the measuring tube.

Before a sample of gas is taken into the apparatus for analysis certain adjustments must be made. In the first place, the reagents in the pipettes must all be brought to a standard level at some arbitrary point, usually indicated by a scratch on the glass tube just below the short rubber tube connecting it to the manifold yoke. This adjustment is

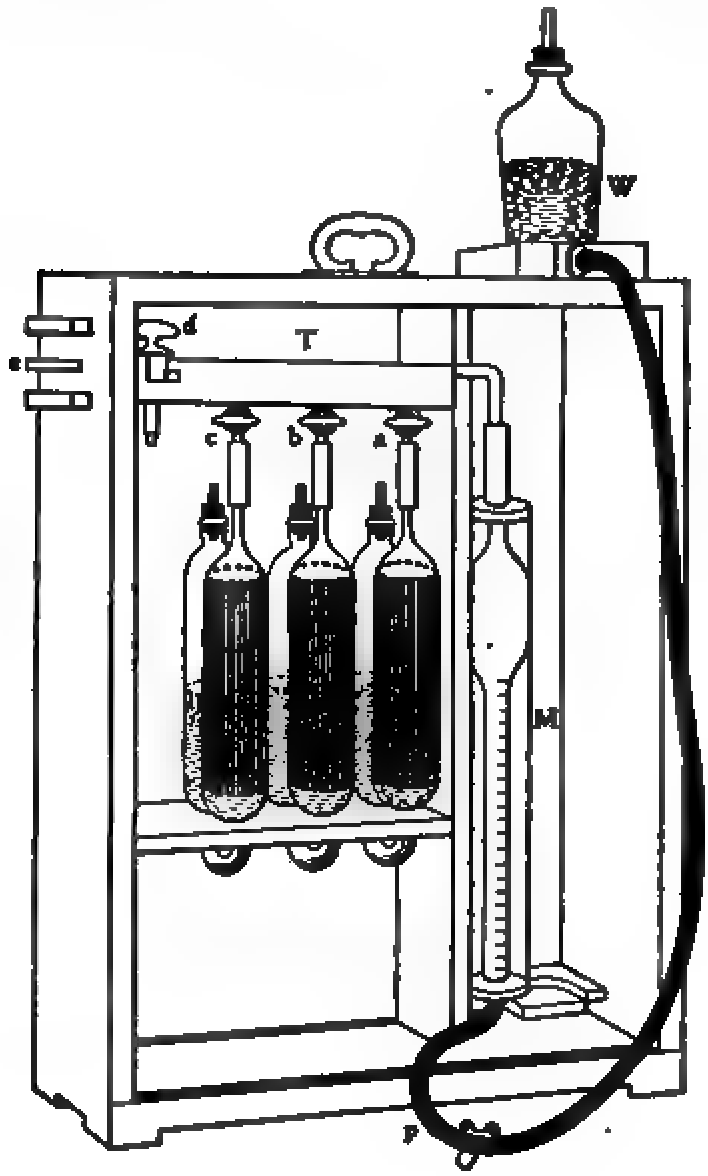


FIG. 281. — Fisher's "Orsat" Apparatus.

FIG. 282. — Pipette of Fisher's "Orsat" Apparatus.

accomplished by opening, one at a time, the valves at the tops of the pipette and removing the air (or the gas as the case may be) from it by lowering the water level in the measuring tube. The position of the water bottle determines, of course, the level of the water in the measuring tube. When all the air and gases remaining from a previous test have been expelled from the apparatus by filling the measuring tube and the tubes comprising the yoke with water, one of the tubes in the sampling bottle should be connected to the apparatus at **e** and by opening the

valve in the yoke at that end and lowering gradually the level of the water in the measuring tube **M**, a sample is obtained for analysis. This sample must be measured by the scale on the measuring tube at atmospheric pressure[1] to the nearest tenth of a cubic centimeter.[2] After the measurement has been made and recorded if the cock in the tube leading to the absorption pipette containing the reagent for absorbing CO_2 is opened and then the water bottle is raised, all of the measured sample of gas can be forced over into the pipette. The reagent acts more rapidly on the gas if the water bottle is raised and lowered a few times. This movement of the water in the measuring tube agitates the gas and also the reagent and exposes more of the gas to the direct action of the absorbent. To increase the surface over which the reagents can act, the pipettes are filled with small glass tubes. When the gas has been in the first pipette for about a minute, it should be drawn back into the measuring tube with the level of the reagent brought back to the mark where it was originally, and the cock should be closed. The pressure of the gas is again made atmospheric and its volume measured. Now repeat this operation until **two or three measurements are obtained which are alike,** showing that all the CO_2 has been absorbed. Then the cock on the tube leading to the pipette containing the absorbent for oxygen can be opened, the gas forced over, and measured several times until a constant volume is observed. Finally the gas is passed into the third pipette for absorbing CO, repeating the operation of measuring as with the other pipettes.

The absorption of oxygen will usually require considerably more time than for the determinations of the carbonic acid (CO_2) and the carbon-monoxide (CO), so that it is unnecessary to make a measurement of the gas until after the gas has been exposed for about three minutes to the reagent. Soft rubber bulbs or bags (see **Fig. 282**) should be attached by means of glass tubes to the corks shown in the pipettes on the farther side in **Fig. 281** and are provided to protect the reagents from absorbing oxygen from the air. Both pyrogallic acid and cuprous chloride will absorb oxygen from atmospheric air, so that the access of fresh air must be prevented. The rubber bags are useful also for the purpose of producing alternately, with the pressure of the hand, suction and pressure for agitating the reagents.

Allen-Moyer Gas Apparatus. A form of flue gas apparatus particularly suitable for portable use, and in which renewals of broken parts can be cheaply and easily made, is illustrated in **Fig. 283.** This appa-

[1] The pressure of the gas is "atmospheric" when the water bottle is held so that the water in it and that in the measuring tube are at the same level.

[2] The scales of practically all measuring tubes used for gas analysis apparatus are graduated in cubic centimeters.

ratus, designed by Professor John R. Allen and the author,[1] is also particularly suitable for the use of engineers because the pipettes containing the reagents can be removed from the apparatus very easily for changing solutions. They can be emptied, refilled, and replaced in a very short time. The absorption pipettes (Fig. 284) are made simply of two glass test-tubes, the smaller one inside the larger one. The small test-tube is held inverted, and has a very small glass "capillary" tube fused into its closed end. The outer tube is closed at the top by a rubber stopper through which the capillary portion of the inner tube passes. Very small glass tubes are placed in the inner test-tube to increase the

Fig. 283. — Allen-Moyer Gas Apparatus.

Fig. 284. — Absorption Pipette of Allen-Moyer Gas Apparatus.

surface for the action of the reagent. The complete pipette is held in place by means of a hard-rubber disk supported on brass screws. The level of the reagent in the pipette is established when the air in the inner tube is drawn out and the level of the liquid rises to a mark on the glass capillary tubes. In the usual forms of the Orsat apparatus the pipettes invariably become leaky at the stopper provided for emptying. In the Allen-Moyer apparatus there is no opportunity for such leakage.

When the sample of the gas is passed through the capillary tube into the inner test-tube, the reagent is displaced and raises the level in the outer test-tube. Similarly when the gas is passed back into the measuring tube the level falls in the outer tube, rises in the inner one, and is

[1] Made by Bausch & Lomb Co., Rochester, N. Y.

brought back to the original level at the mark on the capillary tube. Otherwise the method of operation is the same as described for Fischer's apparatus (**Fig. 281**).

In this apparatus the measuring tube **M** and water bottle **W** are of the conventional type. The yoke is also similar, although usually made of hard rubber to avoid breaking it in transportation. It has also spring pinchcocks instead of ground-glass cocks. When glass cocks are used by inexperienced persons all sorts of difficulties are likely to result, as it often happens that they are not pressed into their seats tightly enough to prevent the loss of gas or the entrance of air. Sometimes the glass cocks will be put into their seats so tightly that it is impossible to move them without breaking. These difficulties, although met often enough in laboratory work, are still more frequently observed in practice.

It sometimes happens that, when **a**, **b** or **c** are open and the pinchcock on the tube between **M** and **W** is closed, the reagent in **A**, **B** or **C** fails, due not to a leak, as is usually supposed, but to the weight of the column of the reagent expanding the gas.

In case any of the reagent in **A** or **B** be drawn over into the measuring tube and into the water, the analysis is not spoiled but may be continued by flushing out the tubes with water through **d** or **e**, or the addition of a little hydrochloric acid to the water in **W** will neutralize the hydrate or pyrogallate and the washing may be postponed until convenient.

To remove pipettes **A**, **B** or **C** when necessary to renew the reagents, disconnect the gas bags and the rubber tube which connects the glass capillary and rubber capillary tubes, loosen the supporting screw and lift the pipette out. The rubber stopper may now be removed and solutions changed.

Gases should be cooled well in the sampling bottle before beginning the analysis, because such gases change $\frac{1}{491}$ of their volume for a variation of one degree Fahrenheit, or a change of 1 per cent in volume for 4.91 degrees. As an example, if the actual percentage of CO_2 is 10, and during the time required for analysis the temperature changed 4.91 degrees Fahrenheit, then there will have been a shrinkage of a volume of one per cent due to temperature, and the apparent volume of CO_2 will be eleven instead of ten.

Producer Gas Analysis. For the analysis of producer and city illuminating gases which are more complex than the flue gases from coal furnaces the Hempel apparatus is generally used. Typical parts of the apparatus are shown in **Figs. 285** and **286**. It is slightly more difficult to operate and must be handled with greater care than a simpler portable apparatus arranged for flue gas analysis.

Essential parts of the apparatus are shown in **Fig. 285**. They are the leveling tube **L**, a measuring burette **B**, and an absorption pipette **P**.

In the operation of the apparatus the pinchcocks C_1 and C_2 are opened and water which has been thoroughly saturated with the kind of gas to be analyzed is poured into the leveling tube until both tubes are about half full. Now raise the leveling tube **L** so that the water from it flows into the burette **B**, making it entirely full. The pinchcock C_1 should now be closed and connect the rubber capillary tubing at the pinchcock to the pipe from which the gas for analysis is to be taken. After connecting to the gas pipe again open the pinchcock C_1 and draw a little more than 100 cubic centimeters of gas into the burette. Allow the apparatus to stand a minute in this position to permit the water clinging to the sides of the burette to drain. Now close the pinchcock C_1 and by raising the leveling tube compress the gas in the burette until the meniscus stands at the 100 cubic centimeter mark, close the pinchcock C_2 on the lower length of rubber tubing and open the pinchcock C_1 at the top of the burette momentarily to release the pressure in it. During this adjustment hold the leveling tube so that the surface of the water in it is on the same level as in the burette. There will be exactly 100 cubic centimeters of gas in the burette at atmospheric pressure if, when the pinchcock C_1 is opened, the meniscus remains at the 100 mark. If, however, the meniscus shifts from the 100 mark, the adjustment will have to be repeated.

Fig. 285. — Hempel Gas Apparatus.

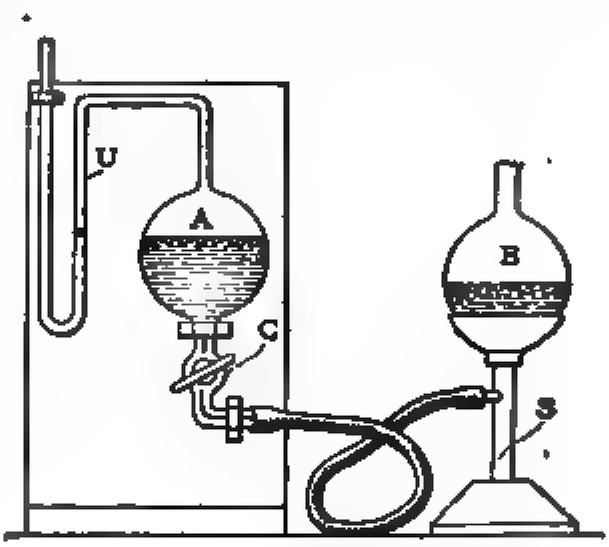


Fig. 286. — Explosion Pipette.

Constituents of the gas must be absorbed in the following order: (1) carbon dioxide (CO_2) with potassium hydrate (KOH)[1] or sodium hydrate (NaOH); (2) "illuminant hydrocarbons" (ethylene C_2H_2, and benzine C_6H_6 in combination) with saturated bromine water[2] or fuming sulphuric

[1] These reagents are the same as used in the flue gas apparatus.

[2] Power Test Committee of A.S.M.E. specified bromine water for the hydrocarbons and NaOH for CO_2. NaOH is cheaper than KOH but the latter is the more rapid absorbent.

acid; (3) oxygen (O_2) with caustic pyrogallic acid; (4) carbon monoxide (CO) with cuprous chloride; (5) marsh gas or methane (CH_4), hydrogen (H_2) and nitrogen (N_2).

A pipette must be provided for each of the reagents and those for pyrogallic acid and cuprous chloride which absorb oxygen must be provided with water seals. Both ends of the pipette for fuming sulphuric acid must be kept closed except when an absorption is being made.

After the sample has been collected in the burette the latter is attached to an absorption pipette as shown in **Fig. 285** by a short piece of bent glass capillary tubing. Before this attachment is made the glass capillary should be filled with water by means of a medicine dropper, so as to avoid the error of entrapping air in this tubing. The pinchcock C_1 should now be open and by raising the leveling tube all the gas should be forced over into the pipette until the water from the burette fills the glass capillary connecting tube. Now close the pinchcock C_1 and shake the pipette lightly to give the gas the best sort of contact with the absorption reagent. After shaking for two or three minutes the gas should be drawn back into the burette by lowering the leveling tube until the solution from the pipette fills the connecting capillary tube when the pinchcock C_1 should be closed. The surfaces of the water in the leveling tube and in the burette should now be brought to the same level. The reading on the scale observed after draining the burette for two minutes as read at the bottom of meniscus should be recorded. The process must be repeated as in the operation of an apparatus for flue gas analysis until the reading is constant. In this way by the use of the proper pipettes all the constituents of the gas to be analyzed, with the exception of H_2 and CH_4, are determined.

Explosion Tests. For determining hydrogen and marsh gas (CH_4) combustion tests are made in a pipette like **Fig. 286** which is made particularly strong for exploding gases over mercury. In the upper portion of the pipette **A** there are two very fine platinum wires which are fused into the glass from opposite sides. When the spark from an induction coil is made to pass through the air gap between the two wires the explosive mixture contained is ignited. The explosion takes place with considerable force so that extreme precautions should be taken to observe that the glass cock **C** and a very strong screw type of pinchcock at the end of the capillary tube **U** are very firmly closed. It is also very desirable to hold a wire screen about a foot square between the operator and the pipette when the spark is made and the explosion occurs.

Procedure for the explosion pipette is as follows: Measure about 15 cubic centimeters (call this **m**) of the **n** cubic centimeters of gas remaining after the absorption of CO by the cuprous chloride and put

the gas remaining in the burette into the pipette for absorbing CO. Now transfer these 15 cubic centimeters to the explosion pipette and immediately thereafter measure about 85 cubic centimeters of air and add this to the gas in the explosion pipette. Then after closing both cocks very carefully and tightly and shaking lightly to insure a good mixture, close the electric circuit momentarily through the induction coil to cause the explosion. After the explosion, and allowing a little time for cooling, measure the gas in the burette. Then pass the same gas into the CO_2 pipette, measure the absorption of CO_2 in the burette, call this **V** cubic centimeters, and finally pass the remainder into the O_2 pipette and determine the volume of oxygen remaining.

In the explosion the following reactions took place:

$$2H_2 + O_2 = 2H_2O.$$
$$CH_4 + 2O_2 = CO_2 + 2H_2O.$$

If **y** is the number of cubic centimeters of oxygen absorbed after the explosion, then since there are 20.8 per cent of oxygen by volume in air, and if **x** cubic centimeters of air were used to make the explosive mixture, the oxygen supplied is 0.208 **x**, and the oxygen used in the explosion is 0.208 **x** − **y**. Assuming the water is all condensed and has practically no volume, the equations above show as regards volumes,

$$\text{Contraction of volume of gas } (z) = \tfrac{3}{2}\,\mathbf{H_2} + 2\,\mathbf{CH_4}. \quad . \quad . \quad . \qquad (85)$$
$$\text{Oxygen used } (0.208\,\mathbf{x} - \mathbf{y}) = \tfrac{1}{2}\,\mathbf{H_2} + 2\,\mathbf{CH_4}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (86)$$
$$CO_2 \text{ formed } (\mathbf{v}) = \mathbf{CH_4}.$$

Subtracting equation (86) from (85),

$$\mathbf{z} - [0.208\,\mathbf{x} - \mathbf{y}] = \mathbf{H_2} \text{ (in } \mathbf{m} \text{ cu. cm. of “ residual ” gas)},$$

and total $\mathbf{H_2}$ in original sample $= \dfrac{\mathbf{n}}{\mathbf{m}}\{\mathbf{z} - [0.208\,\mathbf{x} - \mathbf{y}]\}$.

Similarly CH_4 in **m** cubic centimeters of “ residual ” gas is the absorption of CO_2 (**v**) and the total CH_4 in original sample is $\dfrac{\mathbf{nv}}{\mathbf{m}}$.

Volume of nitrogen **N** can then be obtained by subtracting the sum of all the constituents now determined from the original volume of the sample of gas. Calculation of nitrogen content is, however, a good check on the accuracy of the analysis. Thus if, as above, **x** cubic centimeters of air were mixed with the **m** cubic centimeters of “ residual ” gas for the explosion test, then 0.208 **x** was oxygen and (**x** − 0.208 **x**) was nitrogen. After the CO_2 and O_2 were absorbed from the products of the explosion there were say **w** cubic centimeters of nitrogen remaining, so that the **m** cubic centimeters of “residual ” gas contained **w** − (**x** − 0.208 **x**) of nitrogen. Nitrogen content of the original sample is therefore $\dfrac{\mathbf{n}}{\mathbf{m}}[\mathbf{w} - \mathbf{x}(1 - 0.208)]$.

Results of the analysis should be tabulated and the accuracy as regards the " nitrogen check " stated clearly.

Coefficient of Dilution. The coefficient of dilution is the ratio of the volume of the air supplied to the volume theoretically necessary to provide the oxygen required for combustion. It will now be shown how this coefficient can be calculated from an analysis of the flue gases.

Oxygen when combining with carbon to form carbon dioxide produces a volume equal to itself, thus,

$$C + O_2 = CO_2, \qquad (74)$$

and in forming carbon monoxide produces twice the volume

$$2\,C + O_2 = 2\,CO. \qquad (75)$$

Now if we use symbols to designate the percentages by volume of the gases in a sample of flue gas as follows:

- **a** is the percentage by volume CO_2,
- **b** is the percentage by volume O_2,
- **c** is the percentage by volume CO,
- **d** is the percentage by volume N_2 (nitrogen).

Then the volume occupied by the free oxygen in the air before combining with the carbon was $a + b + \frac{1}{2}c$ per cent, while that required for **complete** combustion is obviously $a + c$ per cent.

The coefficient of dilution[1] is therefore,

$$\frac{a + b + \frac{1}{2}c}{a + c}. \qquad (76)$$

[1] This coefficient is variously defined so that in stating a result the method of computation should be given. Many engineers write it thus:

$$x = \frac{a + b + \frac{1}{2}c}{a + \frac{1}{2}c},$$

where the denominator as given is the air required for the kind of combustion as indicated by the analysis.

Another method sometimes used is based on the nitrogen content. Practically all the nitrogen indicated by the analysis is due to the total air supplied, and using d for the percentage of nitrogen we can write for the nitrogen which was a part of that used for combustion, approximately,

$$d - \tfrac{79}{21}(b - \tfrac{1}{2}c),$$

and

$$x = \frac{d}{d - \frac{79}{21}(b - \frac{1}{2}c)}.$$

This last formula is not as accurate as any of the others, giving usually results about 10 per cent too low. A better method than the last, based on the fact that the air supplied contains 20.8 per cent by volume of oxygen, is stated thus,

$$x = \frac{20.8}{20.8 - b}.$$

In a little different form the reactions given in the last paragraph may be stated (1) for carbon burned to CO_2,

$$\underset{\substack{\text{(solid)}\\24}}{2\,C} + \underset{\substack{\text{(2 vols.)}\\64}}{2\,O_2} = \underset{\substack{\text{(2 vols.)}\\88}}{2\,CO_2}, \quad \ldots \ldots \quad (77)$$

and (2) for carbon burned to CO,

$$\underset{\substack{\text{(solid)}\\24}}{2\,C} + \underset{\substack{\text{(1 vol.)}\\32}}{O_2} = \underset{\substack{\text{(2 vols.)}\\56}}{2\,CO}. \quad \ldots \ldots \quad (78)$$

The ratio of the theoretical volume of the carbon burned to CO_2, to the volume burned to CO is the same as the ratio of the volume of CO_2 in the products of combustion is to the volume of CO. Further since the ratio of volumes of the carbon **vapor** is obviously the same as the ratio of the corresponding weights, we may say that in a mixture of gases the ratio of the **weight** of carbon required to produce the CO_2 to the **weight** needed for the CO is equal to the ratio of the **volume** of CO_2 in the mixture to the **volume** of CO.

The atomic weights of carbon and oxygen show (equation (77)) that the volume of oxygen is $2\frac{2}{3}$ (64/24) times as heavy as an equal volume of carbon vapor. It follows then that for burning one pound of carbon to CO_2, $2\frac{2}{3}$ pounds of oxygen are required. The other reaction (78) showing the combination of carbon and oxygen to form CO shows with the same reasoning that $1\frac{1}{3}$ pounds of oxygen are required to burn one pound of carbon to CO.

In the general case we are considering and using symbols **a** and **c** respectively as before to represent the volumes of CO_2 and of CO in the flue gases, then the weight of the carbon burned to CO_2 is to the weight burned to CO as **a** is to **c** and if $\frac{a}{a+c}$ represents the weight of carbon burned to CO_2 and $\frac{c}{a+c}$ the weight burned to CO, then the weight of oxygen required per pound of carbon is $2\frac{2}{3}\left(\frac{a}{a+c}\right) + 1\frac{1}{3}\left(\frac{c}{a+c}\right)$ and the weight of **air**[1] per pound of carbon is in pounds,

$$\frac{100}{23}\left[2\tfrac{2}{3}\left(\frac{a}{a+c}\right) + 1\tfrac{1}{3}\left(\frac{c}{a+c}\right)\right]. \quad \ldots \ldots \quad (79)$$

If **z** is the percentage by weight of carbon in the coal then the weight of air in pounds per pound of coal is

$$\frac{100\,z}{23}\left[2\tfrac{2}{3}\left(\frac{a}{a+c}\right) + 1\tfrac{1}{3}\left(\frac{c}{a+c}\right)\right]. \quad \ldots \ldots \quad (80)$$

[1] Weight of air may be checked with Peabody's and Jacobus' equations (86, 87, and 88), pages 281 and 283.

Volumetric analyses of the flue gases can be used also to calculate the weight of the products of combustion (flue gases) per pound of coal burned and also the heat units lost in these gases. For this calculation the relations of the molecular weights are important.

Molecular weight of $CO_2 = 44$; $O_2 = 32$; $CO = 28$; $N_2 = 28$; $C = 12$.

Now in a sample of **x** pounds of flue gases in which the percentages by volume are represented by the symbols **a, b, c, d,** the relative percentages by weights of the constituents will be

$CO_2 = \frac{44\,a}{x}$; $O_2 = \frac{32\,b}{x}$; $CO = \frac{28\,c}{x}$; $N_2 = \frac{28\,d}{x}$; and we can write further

Weight of carbon burned to CO_2 in x pounds of gas $= \frac{12}{44} \times 44\,a = 12\,a.$
Weight of carbon burned to CO in x pounds of gas $= \frac{12}{28} \times 28\,c = 12\,c.$
Total weight of carbon burned in x pounds of gas $= 12\,(a + c).$
Total weight of carbon burned per pound of gas $= 12\,\frac{(a+c)}{x}.$
Total weight of gas generated per pound of carbon $= \frac{x}{12\,(a+c)}.$

Of this total weight of gas as expressed by the last equation the constituents are distributed in percentages by weight as follows:

Weight of CO_2 in samples per pound carbon burned,

$$w_1 = \frac{44\,ax}{12\,x\,(a+c)},$$

or we may write

$$CO_2 = w_1 = \frac{44\,a}{12\,(a+c)}\,\text{lb.};$$

and similarly,

$$O_2 = w_2 = \frac{32\,b}{12\,(a+c)}\,\text{lb.};$$

$$CO = w_3 = \frac{28\,c}{12\,(a+c)}\,\text{lb.};$$

$$N_2 = w_4 = \frac{28\,d}{12\,(a+c)}\,\text{lb.}$$

Total weight of gases w_g per pound of coal burned, if there is z **per cent**[1] of carbon in the coal, is in pounds

$$w_g = \frac{z\,(44\,a + 32\,b + 28\,c + 28\,d)}{12\,(a+c)\,100}. \quad \quad (81)$$

Now if we represent by t_f and t_a the temperatures respectively in degrees Fahrenheit of the gases in the flue and of the air entering the

[1] It may be assumed for very approximate values that $z = 1 - y$ where y is the per cent of ash and moisture in the coal.

furnace, then the **heat lost** in the flue gases Q_g per pound of coal is, inserting values of specific heats,[1]

$$Q_g = \frac{z}{100}(.217\,w_1 + .217\,w_2 + .245\,w_3 + .244\,w_4)(t_f - t_a).$$

The total heat generated Q_0 by the more or less incomplete combustion of one pound of coal when there are a and c percentages by volume respectively of CO_2 and CO in the flue gas is

$$Q_0 = \frac{z}{100}\left(\frac{a}{a+c} \times 14{,}600 + \frac{c}{a+c} \times 4400\right)(\text{B.t.u.}), \quad . \quad . \quad (82)$$

since the heat of combustion of carbon when burned to CO_2 is approximately 14,600 and when burned to CO is about 4400 B.t.u.

Finally, if Q_p is the heat from perfect combustion or the "calorific" value in B.t.u. of a pound of coal, then the **efficiency of the furnace**[2] = $\frac{Q_0}{Q_p}$. The percentage of heat from perfect combustion lost in the flue gases $= \frac{Q_g}{Q_p}$.

Recording Apparatus for Determining CO_2. A typical apparatus for making a continuous record of the percentage by volume of carbon dioxide in gases is shown in **Fig. 287.**[3] The gas is taken to the instrument from the side flue or last combustion chamber of each boiler or furnace to the inlet pipe D and is drawn through the machine by a special water aspirator Q, fixed to the top of the instrument by means of the standard T. After actuating the aspirator Q, a portion of the water flows to the small tank L, which serves as a pressure regulator, and is provided with an overflow tube R. From this tank the water enters the tube H in a fine stream, which is adjusted by the cock S and gradually fills the vessel K. This vessel consists of an upper and a lower compartment, the two being in communication through a tube erected in the upper chamber and reaching nearly to the top. Water, which enters this vessel K through the tube H, gradually fills the upper chamber and thus compresses the air contained in it. This pressure is transmitted to the lower compartment through the communication tube mentioned above, and acts upon the mixture of glycerine and water with which this is filled, driving it out into the calibrated tube C. When the rising liquid in C has reached the inlet and outlet to this vessel, no more gas can enter

[1] This method of finding the heat escaping in the flue gases may be used to correct determinations made with the Junkers calorimeter (page 224) when the products of combustion are discharged at a temperature different from that of the room.

[2] For the calculation of related quantities see "Heat Balance" (A.S.M.E. Rules), pages 280 and 281.

[3] Sarco Engineering Co., West Street, N. Y.

the calibrated tube and the aspirator will now draw the gas through the seal *F*.

Before the liquid can close the central tube in *C*, the gas must overcome the slight resistance offered by the elastic bag *P*, and is thereby forced to assume atmospheric pressure. When the liquid has sealed the lower open end of this central tube, exactly 100 cubic centimeters of flue gas are trapped off in the outer vessel *C* and its companion tube, under atmospheric pressure. As the liquid rises, the gas is forced through the thin tube *Z* into the vessel *A*, which is filled with a solution of caustic potash for absorbing carbon dioxide.

The gas remaining gradually displaces the potash solution in *A*, sending it up into the vessel *B*. This has an outer jacket filled with glycerine and supports a float *N*. Through the center of this float reaches a thin tube, through which the air in *B* is kept at atmospheric pressure. The float is suspended from the pen gear *M* by a silk cord and counterbalanced by the weight *X*. The liquid in *B* forces a portion of the air through the central tube in the float, and then raises the latter, causing the pen lever to swing upward, carrying the pen *Y* with it.

Fig. 287. — Recording CO_2 Apparatus.

The mechanism is so calibrated and adjusted that the pen will travel to the top, or zero line, on the chart when only atmospheric air is passing through the machine, and nothing is absorbed by the potash in *A*. When there is any carbon dioxide in the gas it is absorbed by the potash in *A*, and not so much of this liquid would be forced up into the vessel *B*. The float would not then cause the pen to travel up so high on the chart, in proportion to the amount of CO_2 absorbed.

"Precision" Simmance-Abady CO_2 Recorder[1] **(Fig. 288)** is a most satisfactory instrument, being at the same time simple and accurate. Through the valve **V** a continuous flow of water is maintained into the chamber **k**, with an overflow through **o**. Some of this water flows through **E** into the tank **A** and the float **F** rises with the water level. As it rises the cylinder **d** falls since they are joined by a cord **c**. When **F** is at the top of its stroke it raises a valve stem **S**, trips the valve and

[1] Precision Instrument Co., Detroit.

causes the water in **A** to be siphoned out through the tube **g**. This lowering of the water level permits the float **F** to be lowered and at the same time raises the cylinder **d**, making a partial vacuum under it. Chimney gases are thus drawn from the flues into this bell through a supply pipe **P**. The water discharged from the tube **g** into the cup **U** when it overcomes the counterweight **W**, closes the valve **h** in the pipe **P**, and entraps a fixed volume of gas below **d**. In the meantime water has been continuously flowing into **A**, causing **F** to rise again and **d** to drop as before. As **d** goes down the entrapped flue gas is forced by means of the small pipes shown through the KOH solution in **M** and then into the "recorder" chamber **R** which is also water sealed by a cylinder **j**. Displacement of the cylinder **j** will be less in proportion to the volume of gas (CO_2) absorbed. The elevation to which **j** rises is indicated on a scale **N** graduated in per cent of CO_2, and a very simple recording device not shown in the figure registers on a chart corresponding values. Samples are analyzed and records made every three minutes. A branch of the gas pipe **P** goes to **Q** where it enters a small water aspirator supplied with water from the pipe **V** which is continuously exhausting gas from the flues so that the sample entering the instrument shows its true analysis when it was taken.

FIG. 288. — "Precision" CO_2 Recorder.

Uehling's CO_2 Recorder or "Composimeter"[1] is shown diagrammatically in **Fig. 289**. The gas to be analyzed is drawn through the two apertures at **A** and **B** by a constant suction produced by an aspirator. If these apertures are kept at the same temperature, the suction or partial vacuum in the chamber between the two apertures will remain constant so long as all the gas passes through both apertures; if, however, part of the gas be taken away or absorbed in the space between the apertures the vacuum will increase in proportion to the amount of gas absorbed. It is evident that if a manometer or light vacuum gage be connected with this chamber, the amount of gas absorbed will be indicated by the vacuum reading.

The diagram shows the more important parts of the instrument. Gas is drawn from the last pass or uptake of the boiler by means of the

[1] Uehling Instrument Co., Passaic, N. J.

aspirator through a preliminary filter located at the boiler, then through a second filter on the instrument as shown, and finally it passes through aperture **A**, the absorption chamber, and aperture **B**, to the aspirator, where it leaves the instrument with the exhaust steam.

The CO_2 is completely absorbed by the caustic solution as the gas flows through the absorption chamber located between apertures **A** and **B**. Its volume will be reduced which causes a change in the tension (partial vacuum) of the gas between the two apertures. This tension varies with the percentage of CO_2 contained in the gas, and is indicated by a water column at the instrument, and by a recording vacuum gage graduated to read percentages of CO_2.

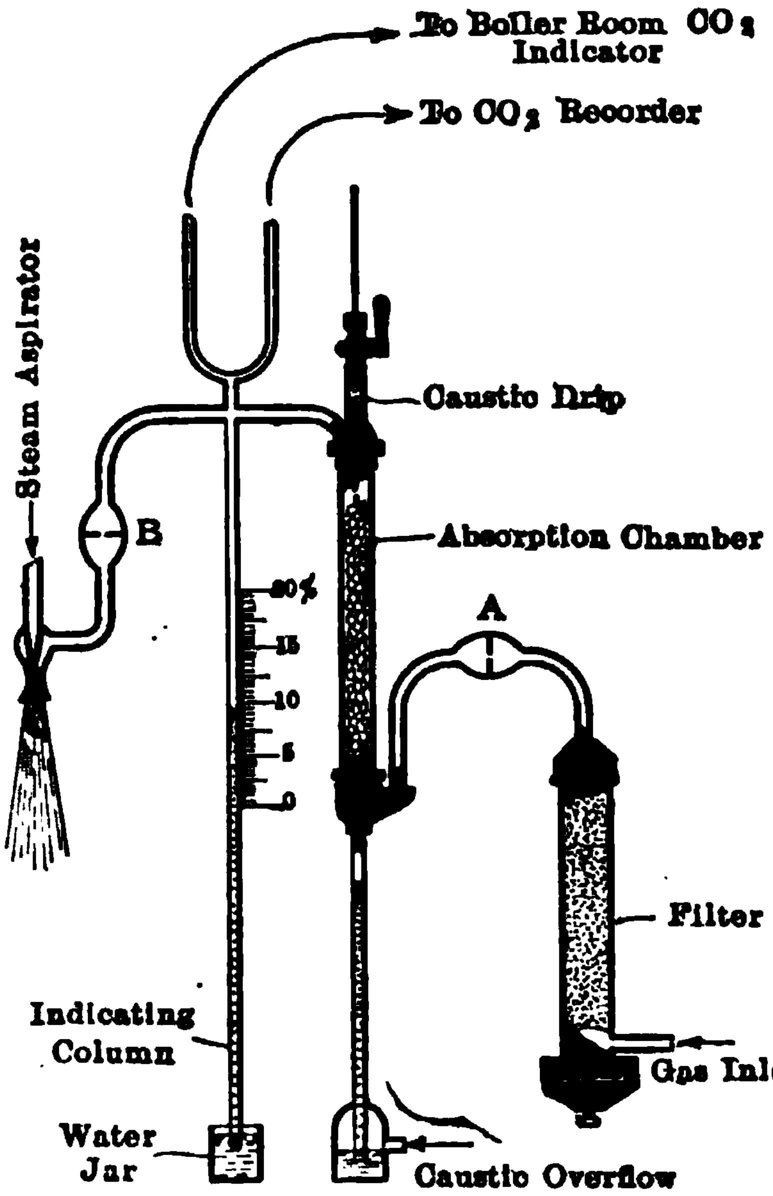


FIG. 289. — Uehling's Gas "Composimeter."

Another apparatus for making continuous determinations of CO_2 in flue gases is shown in **Fig. 290.** Gas from the boiler flue enters at **M**, passes through an excelsior filter where dust is removed, and then goes on through tubes leading it through glass vessels containing cotton wool and calcium chloride. After being cleaned and dried it passes in the direction of the arrows through the valve **B** into the weighing apparatus. On account of the greater specific gravity of CO_2 the larger the percentage of this gas the greater the tendency will be to pull downward the vessel **G** so that the pointer **S** on the balance can be adjusted to make the scale over which it travels indicate the percentage of CO_2. For such a method of determination, obviously, the gas must be clean and dry. The cleaning is done by the excelsior and wool filters and the drying is done by the calcium chloride.

Smoke Determinations. The method most generally used to determine the density of smoke is with a **Ringelmann chart,** which is shown in reduced scale in **Fig. 291.** Cards ruled like those shown, but covering a much larger area, are placed in a horizontal row about 50 feet from the observer and in line with the chimney, together with plain white and black cards. The observer glances rapidly from the chimney to the cards and judges which one corresponds most nearly with the color of the smoke. The lines in cards 1 to 4 are respectively 1, 2.3, 3.7 and 5.5 millimeters thick and the spaces are 9, 7.7, 6.3, and 4.5 millimeters.

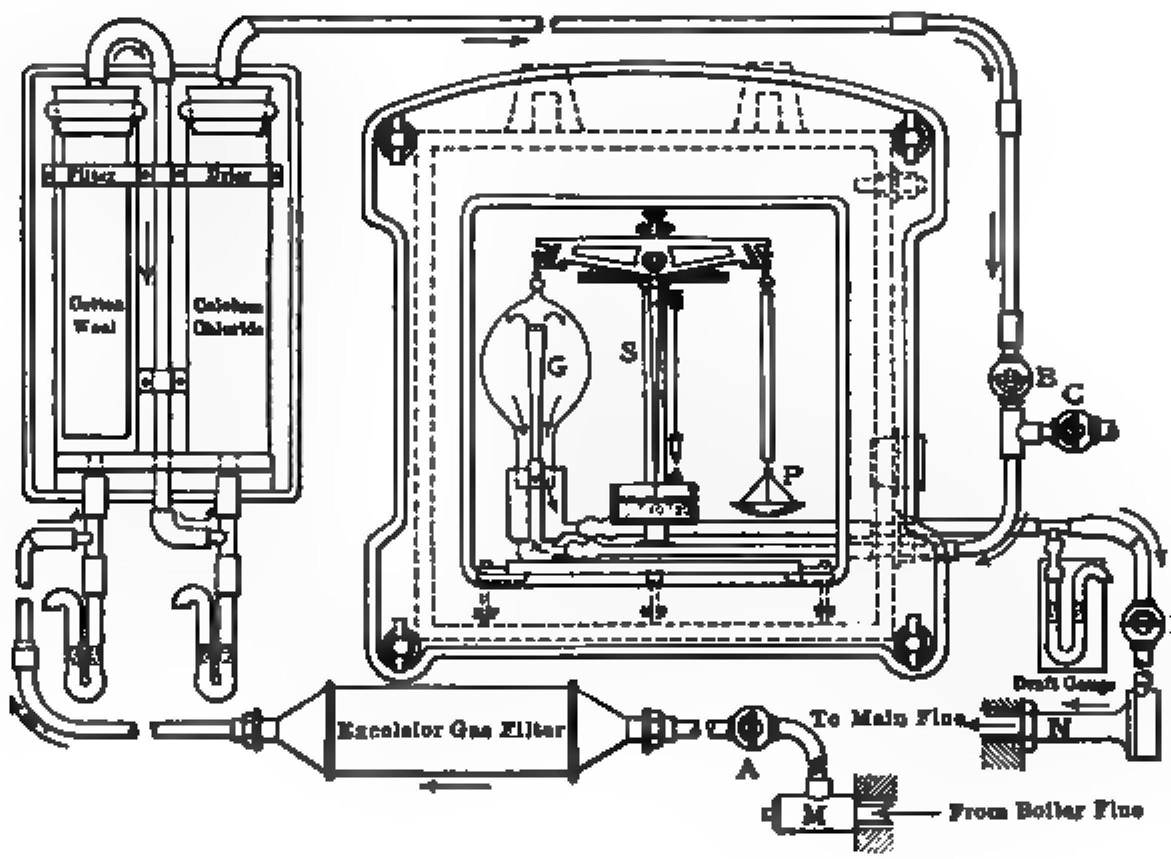


FIG. 290. — CO_2 "Weighing" Apparatus (Econometer).

A soot collecting method is sometimes used. It is applied by suspending by a wire from the top of the flue a plate $\frac{1}{8}$-inch wide and 24 inches long. The hole through which the plate is inserted is kept covered at other times. This plate is temporarily withdrawn every two hours and the collection of soot removed and weighed.

No. 1 No. 2 No. 3 No. 4

FIG. 291. — Ringelmann Smoke Chart.

In another apparatus adopted by the Chicago Commerce Association a continuous sample of gas is drawn from the chimney by means of a special Pitot tube and exhauster. Solid particles in the gas collected are entrapped in a filter. The collecting tube is so arranged that the rate of flow through the apparatus is the same as that through the

chimney, so that when applied to chimneys of different areas the weight of soot, etc., collected is a measure of the density of the smoke.

Eddy Smoke Recorder[1] is one of the few devices for automatically recording the density of smoke. The apparatus consists in its latest form of a steam ejector which draws a continuous flow of gas from the chimney and discharges it through a nozzle against a paper-covered drum revolved by a clock mechanism. The soot in the smoke makes its own record on the chart.

[1] Hamler-Eddy Co., Chicago.

CHAPTER X

INSTRUCTIONS REGARDING REPORTS OF TESTS IN GENERAL[1]

Object. Ascertain the specific object of the test, and *keep this in view* not only in the work of preparation, but also during the progress of the test, and do not let it be obscured by devoting too close attention to matters of minor importance. Whatever the object of the test may be, accuracy and reliability must underlie the work from beginning to end.

If questions of fulfillment of contract are involved, there should be a clear understanding between all the parties, preferably in writing, as to the operating conditions which should obtain during the trial, and as to the methods of testing to be followed, unless these are already expressed in the contract itself.

Among the many objects of performance tests, the following may be noted:

Determination of capacity and efficiency, and how these compare with standard or guaranteed results.

Comparison of different conditions or methods of operation.

Determination of the cause of either inferior or superior results.

Comparison of different kinds of fuel.

Determination of the effect of changes of design or proportion upon capacity or efficiency, etc.

Dimensions. Measure the dimensions of the principal parts of the apparatus to be tested, so far as they bear on the objects in view, or determine these from correct working drawings. Notice the general features of the same, both exterior and interior, and make sketches, if needed, to show unusual points of design.

The dimensions of the **heating surfaces of boilers and superheaters** to be found are those of surfaces in contact with the fire or hot gases.[2] The submerged surfaces in boilers at the mean water level should be considered as **water-heating surfaces,** and other surfaces which are exposed to the gases as **superheating surfaces.**

In the case of condensers, feedwater heaters, and the like, the outside surfaces are to be taken. In reheaters and steam jackets, the surfaces to be considered are those exposed to the steam of lower pressure.

The dimensions of engine cylinders should be taken when they are cold, and,

[1] Report of Committee on Power Tests, *Journal of American Society of Mechanical Engineers*, Nov., 1912.

[2] Heating surface in fire-tube boilers is therefore calculated on the basis of the inside diameter of the tubes.

if extreme accuracy is required, as in scientific investigations, corrections should be applied to conform to the mean working temperature. If the cylinders are much worn, the average diameter should be found. **Clearance of the cylinders** may be determined approximately from working drawings of the engine. For accurate work, when practicable, the clearance should be determined by the **water measurement** method. (See page 293.)

Examination of Plant. Make a thorough examination of the physical condition of all parts of the plant or apparatus which concern the object in view, and record the conditions found, together with any points in the matter of operation which bear thereon.

Boiler Leakage. In boilers, for example, examine for leakage of tubes and riveted or other metal joints. Note the condition of brick furnaces, grates and baffles. Examine brick walls and cleaning doors for air leaks, either by shutting the damper and observing the escaping smoke[1] or by candle-flame test. Determine the condition of heating surfaces with reference to exterior deposits of soot and interior deposits of mud or scale.

See that the steam main or header is so arranged that condensed and entrained water cannot flow back into the boiler.

Ascertain the interior condition of all steam, air, gas, or water cylinders and the condition of their pistons, and of water plungers and impellers, together with the valves and valve-seats belonging thereto. Locate vacuum leaks in exhaust piping, condenser, packings, etc., using vacuum gage or candle-flame test. Examine steam, air, gas, or water piping, traps, drip valves, blow-off cocks, safety valves, relief valves, heaters, etc., and make sure that they do not leak. Determine the condition of the blading, nozzles, and valves in steam turbines, and of buckets, guides and draft-tubes in water turbines.

If the object of the test is to determine the highest efficiency or capacity obtainable, any physical defects, or defects of operation, tending to make the result unfavorable should first be remedied; all fouled parts being cleaned, and the whole put in first-class condition. If, on the other hand, the object is to ascertain the performance under existing conditions, no such preparation is either required or desired.

General Precautions against Leakage. In steam tests make sure that there is no leakage through blow-offs, drips, etc., or any steam or water connections of the plant or apparatus undergoing test, which would in any way affect the results. All such connections should be blanked off, or satisfactory assurance should be obtained that there is leakage neither out nor in. This is a most important matter, and **no assurance should be considered satisfactory** unless it is susceptible of absolute demonstration.

Apparatus and Instruments. Select the apparatus and instruments specified in the Code of Rules[2] applying to the test in hand, locate and install the same, and complete the preparations for the work in view.

[1] Test for air leaks in the setting by firing a few shovelsful of smoky fuel and by immediately closing the damper, observing the escape of smoke through the crevices.

[2] Various codes are given in this book, beginning on page 269.

The arrangement and location of the testing appliances in every case must be left to the judgment and ingenuity of the engineer in charge, the details being largely dependent upon locality and surroundings. One guiding rule, however, should always be kept in view, viz., see that the apparatus and instruments are substantially reliable, and arrange them in such a way as to obtain correct data.

A summary is given below, embracing the entire list of apparatus and instruments referred to in the various Codes, with descriptions of their leading features, methods of application and use, and, where needed, methods of calibration.

(a) *Weighing Scales.* For determining the weight of coal, oil, water, etc., ordinary platform scales serve every purpose. Too much dependence, however, should not be placed upon their reliability without first calibrating them by the use of standard weights, and carefully examining the knife-edges, bearing plates, and ring suspensions, to see that they are all in good order.

Other scales required in connection with test work are small scales for weighing coal-samples used in drying, and laboratory scales for analysis and calorific determinations pertaining to fuels. Such scales should be sensitive to 1/1000 of the quantity weighed.

For testing locomotives and some classes of marine boilers, where room is lacking, sacks or bags are sometimes required to facilitate the handling of coal, the sacks being previously weighed at the time of filling.

(a) *Leakage.*

It is not always necessary to blank off a connecting pipe to make sure that there is no leakage through it. If satisfactory assurance can be had that there is no chance for leakage, this is sufficient. For example, where a straightway valve is used for cutting off a connecting pipe, and this valve has double seats with a hole in the bottom between them, this being provided with a plug or pet cock, assurance of the tightness of the valve when closed can be had by removing the plug or opening the cock. Likewise, if there is an open drip pipe attached to an unused or empty section of pipe beyond the valve, the fact that no water escapes here is sufficient evidence of the tightness of the valve. The main thing is to have positive evidence in regard to the tightness of the connections, such as may be obtained by the means suggested above; but where no positive evidence can be obtained, or where the leakage that occurs cannot be measured, it is of the utmost importance that the connections should be broken and blanked off.

Leakage of relief valves which are not tight, drips from traps, separators, etc., and leakage of tubes in the feed-water heater must all be guarded against or measured and allowed for.

It is well, as an additional precaution, to test the tightness of the feed-water pipes and apparatus concerned in the measurement of the water by running the pump at a slow speed for, say, fifteen minutes, having first shut the feed valves at the boilers and making sure they are tight. Leakage will be revealed by disappearance of water from the supply tank. In making this test a gage should be placed on the pump discharge to guard against undue or dangerous pressure.

(b) *Water-glass Tests of Leakage.*

To determine the leakage of steam and water from a boiler and steam pipes, etc., the water-glass method may be satisfactorily employed. This consists of shutting off

all the feed valves (which must be known to be tight) and the main feed valve, thereby stopping absolutely the entrance or exit of water at the feed pipes to the boiler; then maintaining the steam pressure (by means of a very slow fire) at a fixed point, which is approximately that of the working pressure, and observing the rate at which the water falls in the gage glasses. It is well, in this test, as in other work of this character, to make observations every ten minutes, and to continue them for such length of time that the differences between successive readings attain a constant rate. In many cases the conditions will have become constant at the expiration of fifteen minutes from the time of shutting the valves, and thereafter the fall of water due to leakage of steam and water become approximately constant. It is usually sufficient, after this time, to continue the test for two hours, thereby obtaining a number of half-hourly periods. When this test is finished, the quantity of leakage is ascertained by calculating the volume of water which has disappeared, using the area of the water level and the depth shown on the glass, making due allowance for the weight of one cubic foot of water at the observed pressure. **The water columns should not be blown down during the time a water-glass test is going on, nor for a period of at least one hour before it begins.**

If there is opportunity for condensation to occur and collect in the steam pipe during the leakage test, the quantity should be determined as closely as desirable, and properly allowed for.

(c) *Piston Rod and Valve Rod Leakage.*

In making an engine test where the steam consumption is determined from the amount of water discharged from a surface condenser, leakage of the piston rods and valve rods should be guarded against; for if these are excessive the test is of little use, as the leakage consists partly of steam that has **already done work in the cylinder** and of water condensed from the steam when in contact with the cylinder. If such leakage cannot be prevented, some allowance should be made for the quantity thus lost. The weight of water as shown at the condenser must be increased by the quantity allowed for this leakage.

MISCELLANEOUS INSTRUCTIONS

The person in charge of a test should have the aid of a sufficient number of assistants, so that he may be free to give special attention to any part of the work whenever and wherever it may be required. He should make sure that the instruments and testing apparatus continually give reliable indications, and that the readings are correctly recorded. He should also keep in view, at all points, the operation of the plant or part of the plant under test and see that the operating conditions determined on are maintained and that nothing occurs, either by accident or design, to vitiate the data. This last precaution is especially needed in guarantee tests.

Before a test is undertaken, it is important that the boiler, engine, or other apparatus concerned, shall have been in operation a sufficient length of time to attain working temperatures and proper operating conditions throughout, so that the results of the test may express the true working performance.

It would, for example, be manifestly improper to start a test for determining the maximum efficiency of an externally fired boiler with brick setting, until the boiler had been at work a sufficient number of days to dry out thoroughly and heat the brick work to its working temperature; and likewise improper to begin an engine test for determining the performance under certain prearranged conditions until those conditions had become established by a suitable preliminary run.

An exception should be noted where the object of the test is to obtain the working performance, including the effect of preliminary heating in which case all the conditions should conform to those of regular service.

In preparation for a test to demonstrate maximum efficiency, it is desirable to run **preliminary tests** for the purpose of determining the most advantageous conditions.

In all tests in which the object is to determine the performance under conditions of maximum efficiency, or where it is desired to ascertain the effect of predetermined conditions of operation, all such conditions which have an appreciable effect upon the efficiency should be maintained as nearly uniform during the trial as the limitations of practical work will permit. In a stationary steam plant, for example, where maximum efficiency is the object in view, there should be uniformity in such matters as steam pressure, times of firing, quantity of coal supplied at each firing, thickness of fire, and in other firing operations; also in the rate of supplying the feed-water, in the load on the engine or turbine, and in the operating conditions throughout. On the other hand, if the object of the test is to determine the performance under working conditions, no attempt at uniformity is either desired or required unless this uniformity corresponds to the regular practice and when this is the object the usual working conditions should prevail throughout the trial.

A log of the data should be entered in notebooks or on blank sheets **suitably prepared in advance.** This should be done in such manner that the test may be divided into hourly periods, or if necessary, periods of less duration, and the leading data obtained for any one or more periods as desired, thereby showing the degree of uniformity obtained.

The readings of the various instruments and apparatus concerned in the test other than those showing quantities of consumption (such as fuel, water, and gas) should be taken at intervals not exceeding half an hour and entered in the log. Whenever the indications fluctuate, the intervals should be reduced according to the extent of the fluctuation. In the case of smoke observations, for example, it is often necessary to take observations every minute, or still oftener, continuing these throughout the period covering the range of variations.

Make a memorandum of every event connected with the progress of a test, however unnecessary at the time it may appear. A record should

be made of the exact time of every such occurrence and the time of taking every weight and every observation. For the purpose of identification the **signature of the observer and the date should be affixed to each log sheet or record.**

In the simple matter of weighing coal by the barrow-load, or weighing water by the tank-full, which is required in many tests, a series of **marks, or tallies, should never be trusted.** The time each load is weighed or emptied should be recorded. The weighing of coal should not be delegated to unreliable assistants, and whenever practicable, one or more men should be assigned solely to this work. The same may be said with regard to the weighing of feedwater.

To show the uniformity of the data at a glance the whole log of the trial should be plotted on a chart, using horizontal distances to represent times of observation, and vertical distances on suitable scales to represent various data as recorded. Such a chart showing the log of a boiler test is illustrated in **Fig. 296,** page 268.

It is very heplful to **plot the leading data on such a chart while the test is in progress.**

Report of a test should present all the leading facts bearing on the design, dimensions, condition, and operation of the apparatus tested, and should include a description of any other apparatus and auxiliaries concerned, together with such sketches as may be needed for a clear understanding of all points under consideration. It should state clearly the object and character of the test, the methods followed, the conditions maintained, and the conclusions reached, closing with a tabular summary of the principal data and results.

The standard units on which to base the various measures of capacity and the standard forms of expressing efficiency and economy to which the codes apply are as follows:

STANDARD UNITS OF CAPACITY

a Boilers[1]	One pound of water evaporated into dry steam from and at 212 deg. F. per hour.	
b Reciprocating Steam Engines	One indicated horse power developed in the main cylinders	i.h.p.
	One brake horse power delivered by the main shaft	b.h.p.
c Steam Turbines	One brake horse power delivered by the main shaft	b.h.p.

[1] A subsidiary unit which may be used for stationary boilers is a "Boiler Horse Power," or 34½ pounds of water evaporated from and at 212 deg. F. per hour, i.e., from water at 212 deg. F. into steam at the same temperature. Electrical engineers have suggested a unit termed "Myriawatt," which differs but little from boiler horse power when expressed in B.t.u. per hour.

d Turbo-generators (including engine-driven generators)	One kilowatt-hour delivered at the busbar,[1] not including exciter output[2].	kw.-hr.
e Pumping Machinery. . .	One gallon of water discharged to the force main in 24 hours.	
	One gallon of water discharged per min [3].	gal. per min.
	One water horse power delivered to the force main, based on the total head including suction.	w.h.p.
f Air Machinery.	One cu. ft. of air at 62 deg. F. and 30-in.[4] barometer	cu. ft.
	One air horse power.	air h.p.
g Locomotives.	One indicated horse power developed in the main cylinders.	i.h.p.
	One dynamometer horse power delivered to the draw-bar.	dyn. h.p.
h Gas Producers.	One pound of dry fuel consumed per hour	
i Gas and Oil Engines. . .	One brake horse power delivered by the main shaft.	b.h.p.
j Waterwheels.	One brake horse power delivered by the main shaft.	b.h.p.

STANDARDS OF EFFICIENCY AND ECONOMY

a Boilers.	Relation between B.t.u. absorbed by boiler per lb. of coal fired and calorific value of 1 lb. coal. (Efficiency of boiler furnace and grate.)
	Relation between B.t.u. absorbed by boiler, per lb. of combustible burned and calorific value of 1 lb. combustible. (Efficiency of boiler and furnace.)
b Reciprocating Steam Engines.	(1) B.t.u. per i.h.p.-hr.
	(2) B.t.u. per brake h.p.-hr.
	(3) Thermal efficiency ratio referred to i.h.p.
	(4) Thermal efficiency ratio referred to b.h.p.
	(5) Lbs. of dry steam per i.h.p.-hr.
	(6) Lbs. of dry steam per b.h.p.-hr.
c Steam Turbines.	(1) B.t.u. per b.h.p.-hr.
	(2) Thermal efficiency ratio.
	(3) Lbs. of dry steam per b.h.p.-hr.
d Turbo-generators, (including engine-driven generators). .	(1) B.t.u. per kw.-hr.
	(2) Thermal efficiency ratio.
	(3) Lbs. of dry steam per kw.-hr.

[1] It is assumed that the drop in voltage between generator terminals and switchboard is not over one-half of one per cent.

[2] If the exciter current is taken from an outside source the kw. thus supplied are to be deducted from the total output.

[3] This unit applies to small pumps and some classes of large-sized pumps.

[4] 30-in. barometer is referred to a temperature of 62 deg. F.; or 29.92-in. referred to 32 deg. F. (see page 223).

e Pumping Engines	(1) Ft.-lbs. of work per million B.t.u. (2) Ft.-lbs. of work per 1000 lbs. dry steam. (3) Ft.-lbs. of work per 100 lbs. dry fuel.
f Air Machinery	(1) B.t.u. per net air h.p. per hr. (2) Lbs. of dry steam per net air h.p. per hr. (3) Lbs. of dry steam per 1000 cu. ft. of free air compressed to 100 lbs. gage pressure reduced to atmospheric temperature.
g Locomotives	(1) Lbs. of dry fuel per i.h.p.-hr. (2) Lbs. of dry fuel per dyn. h.p.-hr. (3) Lbs. of dry steam per i.h.p.-hr. (4) Lbs. of dry steam per dyn. h.p.-hr. (5) Lbs. of dry fuel per ton-mile.
h Gas Producers	Relation between B.t.u. of the gas output per lb. of fuel charged and calorific value of 1 lb. of fuel.
i Gas and Oil Engines	(1) B.t.u. per brake h.p.-hr. (2) Thermal efficiency ratio referred to b.h.p. (3) Lbs. of dry fuel per b. h.p.-hr. (4) Cu. ft. of gas per b. h.p.-hr.
j Waterwheels	Relation between brake h.p. and potential h.p. of total water used.
k Steam Power Plants	(1) Lbs. of dry fuel per i.h.p.-hr., main engine. (2) Lbs. of dry fuel per i.h.p.-hr., auxiliaries. (3) Lbs. of dry fuel per i.h.p.-hr., whole plant. (4) Lbs. of dry steam per i.h.p.-hr., main engine. (5) Lbs. of dry steam per i.h.p.-hr., auxiliaries. (6) Lbs. of dry steam per i.h.p.-hr., whole plant.
l Electric Power Plants	(1) Lbs. of dry fuel per kw.-hr., main engine or turbine. (2) Lbs. of dry fuel per kw.-hr., auxiliaries. (3) Lbs. of dry fuel per kw.-hr., whole plant. (4) Lbs. of dry steam per kw.-hr., main engine or turbine. (5) Lbs. of dry steam per kw.-hr., auxiliaries. (6) Lbs. of dry steam per kw.-hr., whole plant.
m Pumping Engine Plants	(1) Ft.-lbs. of work per million B.t.u. (2) Ft.-lbs. of work per 100 lbs. dry fuel. (3) Ft.-lbs. of work per 1000 lbs. dry steam.
n Gas Power Plants	(1) Lbs. of dry fuel per brake h.p.-hr. (2) Cu. ft. of gas per b.h.p.-hr.

The i.h.p. and b.h.p. in this table refer to that of the main engine, turbine, or waterwheel, and the kw. to the current measured at the busbar, not including exciter current. (See second footnote, page 264.)

Contracts for power plant apparatus should specify the leading dimensions of the apparatus and its rated capacity, expressed in the units given in the table. If a specific guarantee of capacity is made, either working capacity or maximum capacity, the operating conditions under which the guarantee is to be met should be clearly set forth, such, for example, as steam pressure, speed, vacuum, quality of fuel, force of draft, etc. Likewise if a contract contains a guarantee of economy all the conditions should be fully specified.

Commercial Ratings. The commercial rating of capacity determined on for power plant apparatus, whether for the purpose of contracts, for sale, or otherwise, should be such that a **sufficient reserve capacity** beyond the rating is available to meet the contingencies of practical operation; such contingencies, for example, as the loss of steam pressure and capacity due to cleaning fires, inferior coal, oversight of the attendants, sudden demand for an unusual output of steam or power, etc. To secure this end, the following requirements should be met:

(a) *Boilers.* The reserve capacity of a boiler should be at least one-third of the commercial rating, when using coal which is regarded as a standard where the boiler is located, the fire being crowded, and the draft between the damper and the boiler being at least ½-in. water column (draft gage).

A sufficient amount of grate surface should be provided in such a boiler to develop the rated capacity with the coal and draft named without crowding the fire.

(b) *Reciprocating Steam Engines and Steam Turbines.* The reserve capacity of a steam engine or steam turbine at a stipulated steam pressure should be such as to allow a drop of at least 15 per cent in the pressure without material reduction in the normal speed at its rated capacity or load. It should also allow an overload at the specified pressure amounting to at least 25 per cent of the rated power.

(c) *Pumping Engines.* The reserve capacity of a pumping engine should be such as to permit a drop in the steam pressure of at least 15 per cent without sensible reduction in the quantity of water discharged at its rated capacity, and to allow an increase in power sufficient to discharge 20 per cent more water than the rated amount.

(d) *Gas Producers.* The reserve capacity of a gas producer should be such that when forced it will burn in a given time 20 per cent more coal of the quality agreed upon than the rated capacity.

(e) *Gas and Oil Engines.* The reserve capacity of an internal-combustion engine should be such that when supplied with gas of the kind and quality which it is designed to use, it should develop at least 20 per cent more power than the commercial rating.

(f) *Waterwheels.* The reserve capacity of a waterwheel should be at least 10 per cent more than the commercial rating at the specified head, the buckets in the wheel being clean and the flow of water unobstructed.

CHAPTER XI

BOILER TESTING

Tests of steam boilers are made to determine usually the following principal results:

(1) Quantity of steam evaporated or furnished per hour.

(2) Efficiency as a heat user, or weight of water evaporated per pound of combustible (fuel less moisture and ash).

(3) Weight of water evaporated per hour per square foot of water-heating surface.

(4) Weight of fuel burned per hour per square foot of grate surface.

Leakages of any kind are always a lurking enemy for those engaged in any kind of accurate testing, and work with boilers is no exception. Poor results with boilers are due more often to air leakage than to any other fault. Air entering the setting and flues instead of the furnace does not assist combustion, but, on the contrary, absorbs from the hot gases a quantity of heat which otherwise might pass through the boiler-heating surfaces into the water in the boiler. When the object of a series of tests is, for example, to compare one kind of coal with another, or one type of grate or mechanical stoker with another, the losses due to air leakage would not be of much consequence in what are only comparative results; but if a boiler is to give the best possible efficiency and capacity, air leaks must be stopped.

In practice the importance of closing air leaks in the boiler setting is forcefully presented when patented devices for fuel saving are installed in boiler plants. Important economies in many cases are guaranteed if the new device is adopted, and then the claims of the agent are made good by instructing his workmen to go over the boiler, closing up all cracks in the setting through which cold air could enter, and at the same time covering the outside surface of the setting with a coating impervious to air. By such means the owner of the plant pays a high price for results that could have been obtained much more cheaply.

The first of the principal objects of a boiler test stated above is to determine its capacity "rating."[1] The unit of capacity most generally

[1] When the steam supply is small or if, for any reason, the noise due to escaping steam from the calorimeters used is objectionable, a calorimeter may be shut off sometimes between readings. The length of time the calorimeter is in operation must then be carefully noted in order to determine the weight of steam lost through it by cal-

used in steam boiler practice is the "boiler" horse power. Now this term horse power has two very distinct meanings in engineering practice. Usually it is taken to mean the rate of doing work or the work done in a definite period of time. In this sense it means, as in the case of engines, turbines, waterwheels, etc., 33,000 foot-pounds per minute.[1] In the case of a steam boiler, however, where the work done must be measured by the conversion of water into steam, a horse power is taken as the evaporation of 30 pounds of water at a temperature of 100 degrees Fahrenheit into steam at 70 pounds pressure above the atmosphere. When this unit was adopted it was considered that 30 pounds per hour was approximately the requirement per indicated horse power of an

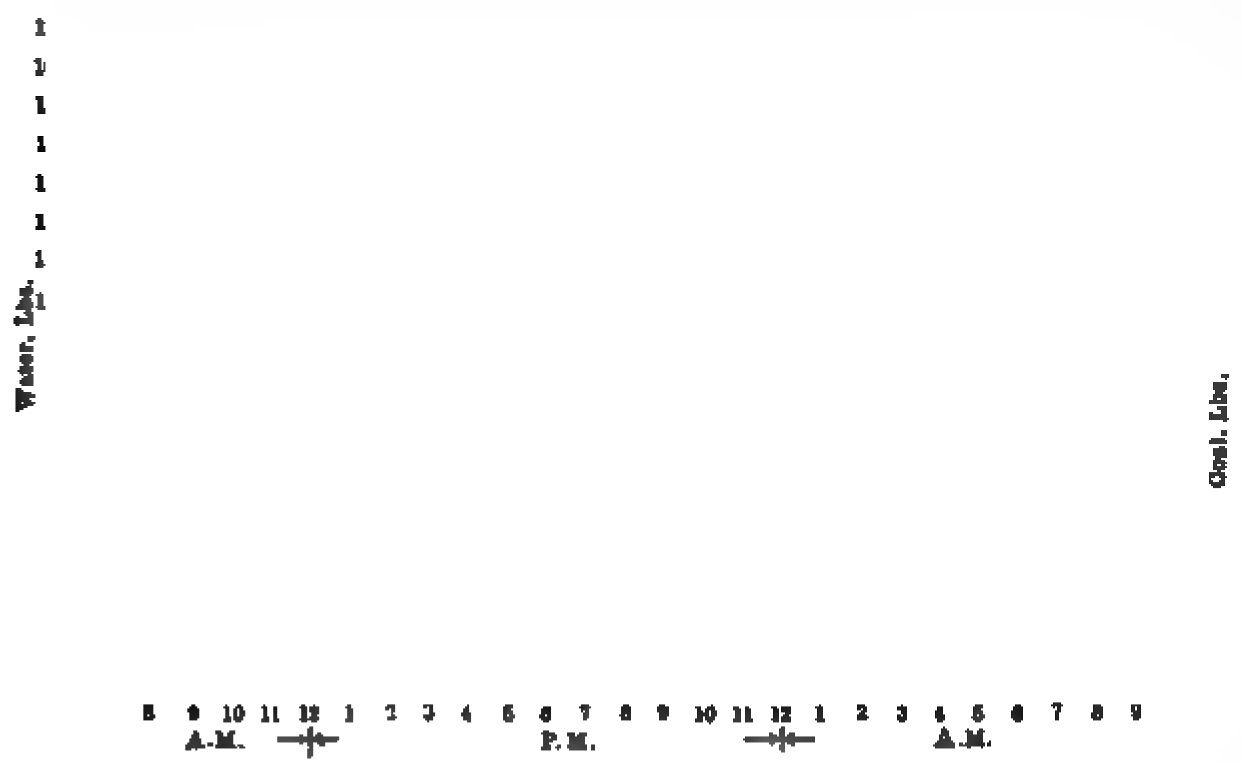


Fig. 296. — Graphical Chart of a Boiler Trial.

average engine. The Committee on Boiler Tests of the American Society of Mechanical Engineers have adopted what is in effect the same unit, stating it, however, somewhat differently — that **a boiler horse power is equivalent to evaporating 34.5 pounds of steam per hour from feed-water temperature of 212 degrees Fahrenheit into steam at the same temperature.** According to the latest steam tables this is equivalent to approximately 33,480 B.t.u. per hour, or 558 B.t.u. per minute.

Unit of Evaporation. For reducing the results of the boiler tests to a common standard the term "unit of evaporation" is used. It is the heat required to evaporate a pound of water from and at 212 degrees

culating the flow through its orifice (see page 189). The flow of steam, however, through the calorimeter must always be started before observations of the temperatures are to be taken in order to get constant conditions.

[1] This unit of horse power was adopted by James Watt, who considered it equivalent to the work done by a good London draft horse.

Fahrenheit,[1] which according to standard steam tables is approximately equivalent to 970.4 B.t.u.[2]

Graphical Log Sheets of boiler tests similar to the one shown in **Fig. 296** are very serviceable for checking the observations when made during the test as the data are taken. In the report of a test it shows also the relative irregularity or regularity of the conditions affecting the results.

Standard Methods for Boiler Trials. The American Society of Mechanical Engineers has adopted rules for conducting boiler trials which are generally accepted in America and are also considered with favor in England.[3] These rules are so complete that they will be given here with practically no abridgment.[4]

RULES FOR CONDUCTING EVAPORATIVE TESTS OF BOILERS

A.S.M.E. CODE OF 1912

General Procedure. Determine the object, take the dimensions, note the physical conditions, examine for leakages, install the testing appliance, etc., as pointed out in the general instructions given on pages 258 to 266, and make preparations for the test accordingly.

Fuel. Determine the character of fuel to be used.[5] For tests of maximum efficiency or capacity of the boiler to compare with other boilers, the coal should be of some kind which is commercially regarded as a standard for the locality where the test is made.

In the Eastern States the standards thus regarded for semi-bituminous coals are Pocahontas (Va. and W. Va.) and New River (W. Va.); for anthracite coals those of the No. 1 buckwheat size, fresh-mined, containing not over 13 per cent ash by analysis; and for bituminous coals, Youghiogheny and Pittsburg coals. In some sections east of the Allegheny Mountains the semi-bituminous Clearfield (Pa.) and Cumberland (Md.) are also considered as standards. These coals when of good quality possess the essentials of excellence, adaptability to various kinds of furnaces, grates, boilers, and methods of firing required, besides being widely distributed and generally accessible in the Eastern market.

There are no special grades of coal mined in the Western States which are widely and generally considered as standards for testing purposes; the best coal obtainable in any particular locality being regarded as the standard of comparison.

A coal selected for maximum efficiency and capacity tests should be the best of its class, and especially free from slagging and unusual clinker-forming impurities.

[1] See also Equivalent Evaporation defined in same units, page 275.

[2] Marks and Davis' *Steam Tables and Diagrams*, see also *Peabody's Steam Tables*.

[3] *Engines and Boilers* by W. W. F. Pullen, pages 466–475.

[4] *Journal of American Society of Mechanical Engineers*, vol. 34, pages 1693–1872.

[5] This code relates primarily to tests made with coal. For reference to oil and gas fuel tests see page 276.

The size of the coal, especially where it is of the anthracite class, should be determined by screening a suitable sample.

Screens for Sizing Coal. The dimensions of screen openings to be used for sizing anthracite coals are given in the following table, the sizes in each case being the opening through which the specified grade will pass, and that over which it will be carried without passing through. The openings referred to are circular.

ANTHRACITE COAL SIZES

Name.	Diameter of Opening through or over which Coal will pass, ins.		Name.	Diameter of Opening through or over which Coal will pass, ins.	
	Through.	Over.		Through.	Over.
Broken	$4\frac{1}{2}$	$3\frac{1}{4}$	No. 1 Buckwheat	$\frac{9}{16}$	$\frac{5}{16}$
Egg	$3\frac{1}{4}$	$2\frac{5}{16}$	No. 2 Buckwheat	$\frac{5}{16}$	$\frac{3}{16}$
Stove	$2\frac{5}{16}$	$1\frac{5}{8}$	No. 3 Buckwheat	$\frac{3}{16}$	$\frac{3}{32}$
Chestnut	$1\frac{5}{8}$	$\frac{7}{8}$	Culm	$\frac{3}{32}$	..
Pea	$\frac{7}{8}$	$\frac{9}{16}$			

The sizes and grades of bituminous and semi-bituminous coals vary so much according to kind and locality that there are no standards of size for these coals which are generally recognized.

Bituminous coals in the Eastern States may be graded and sized as follows:

(*a*) Run of mine coal; the unscreened coal taken from the mine.
(*b*) Lump coal; that which passes over a bar-screen with openings 1¼ in. wide.
(*c*) Nut coal; that which passes through a bar-screen with 1¼-in. openings and over one with ¾-in. openings.
(*d*) Slack coal; that which passes through a bar-screen with ¾-in. openings.

Bituminous coals in the Western States may be graded and sized as follows:

(*a*) Run of mine coal; the unscreened coal taken from the mine.
(*b*) Lump coal; divided into 6-in., 3-in. and 1¼-in. lump, according to the diameter of the circular openings over which the respective grades pass; also 6 by 3 lump and 3 by 1¼ lump, according as the coal passes through a circular opening having the diameter of the larger figure and over that of the smaller diameter.
(*c*) Nut coal; divided into 3-in. steam nut, which passes through an opening 3-in. diameter and over 1¼-in. diameter opening; 1¼-in. nut, which passes through a 1¼-in. diameter opening and over a ¾-in. diameter opening; ¾-in. nut, which passes through a ¾-in. diameter opening and over a ⅝-in. diameter opening.
(*d*) Screenings; that which passes through a 1¼-in. diameter opening.

Apparatus and Instruments. The apparatus and instruments required for boiler tests are:

(*a*) Platform scales for weighing coal and ashes.
(*b*) Graduated scales attached to the water glasses.

(c) Tanks and platform scales for weighing water (or water meters calibrated in place).
(d) Pressure gages, thermometers, and draft gages.
(e) Calorimeters for determining the calorific value of fuel and the quality of steam.
(f) Furnace pyrometers.
(g) Gas analyzing apparatus.

Full directions regarding the use and calibration of the above-mentioned appliances are given in the preceding chapters.

Location of Instruments. (a) The feedwater thermometer should be placed in a thermometer well inserted in the feed pipe. Except **in cases where an injector is used**[1] the point selected should be as near as practicable to the boiler. Where an injector is employed, and the water is weighed or measured before it is supplied thereto, the well should be placed on the suction side of the injector, and the injector should receive steam through a short covered pipe connected directly to the boiler under test. If the steam is taken from some other source and it is of different pressure and different quality from that of the boiler under test, correction should be made for such difference, and especially for any excessive moisture thus introduced into the feedwater. When the temperature of the water changes between the injector and boiler, as by the use of a heater or by excessive radiation, the temperature at which the water not only enters and leaves the injector, but that also at which it enters the boiler, should also be taken. In that case, the weight to be used is that of the water leaving the injector, computed from the heat units if not directly measured and the temperature, that of the water entering the boiler. The weight of condensed steam to be added to the weight of water entering the injector, to obtain that leaving the injector, may be computed by multiplying the weight entering by the proportion

$$\frac{h_3 - h_1}{h_2 - h_3},$$

in which

h_1 = heat units per pound of water entering injector.
h_2 = heat units per pound of steam entering injector.
h_3 = heat units per pound of water leaving injector.

(b) The location of the steam calorimeter and steam thermometer should be as close to the boiler as possible.

(c) **Draft gages** should be attached to each boiler between the hand damper and the boiler, and as near the damper as practicable. In the case of a plant containing a number of boilers, a gage should also be attached to the main flue between the regulating damper and the boiler plant. It is desirable also to have gages connected to the furnace or furnaces of the boilers, and in cases of forced blast, to the ashpits and blower ducts. If there is an economizer in the flue a gage should be connected to the flue at each end of this apparatus. The same draft gage may be used for all the points.

[1] In feeding a boiler undergoing test with an injector taking steam from another boiler, or from the main steam pipe from several boilers, the evaporative results may be modified by a difference in the quality of the steam from such source compared with that supplied by the boiler being tested, and in some cases the connection to the injector may act as a drip for the main steam pipe. If it is known that the steam from the main pipe is of the same pressure and quality as that furnished by the boiler undergoing the test, the steam may be taken from such main pipe.

noted, provided suitable pipes are run from the gage to each, arranged so as to be readily connected to any of these points at will.

(*d*) The flue thermometer should be located where it will show the average temperature of the whole body of gas. For an extremely large flue the thermometer may be placed in an oil pot of small diameter, which is suspended in the flue, and the thermometer lifted partially out of the oil when the temperature is read.

(*e*) **Samples for flue gas analysis** should be drawn from the region near the center of the main body of escaping gases, and the point selected should be one where there is no chance for air leakage into the flue, which could affect the average quality. In a round or square flue having an area of not more than one-eighth of the grate surface, the sampling pipe may be introduced horizontally at a central point, or preferably a little higher than the central point, and the pipe should contain perforations extending the whole length of the part immersed and pointing toward the current of gas, the collective area of the perforations being less than the area of the pipe.

Duration. The duration of tests to determine the efficiency of a hand-fired boiler, should be 10 hours of continuous running, or such time as may be required to burn a total of 250 pounds of coal per square foot of grate.

In the case of a boiler using a mechanical stoker, the duration, where practicable, should be at least 24 hours. If the stoker is of a type that permits the quantity and condition of the fuel bed at beginning and end of the test to be accurately estimated, the duration may be reduced to 10 hours, or such time as may be required to burn the above noted total of 250 pounds per square foot.

In commercial tests where the service requires continuous operation night and day, with frequent shifts of firemen, the duration of the test, whether the boilers are hand-fired or stoker-fired, should be at least 24 hours. Likewise in commercial tests, either of a single boiler or of a plant of several boilers, which operate regularly a certain number of hours and during the balance of the day the fires are banked, the duration should not be less than 24 hours.

The duration of tests to determine the maximum evaporative capacity of a boiler, without determining the efficiency, should not be less than three hours.

Starting and Stopping. The conditions regarding the temperature of the furnace and boiler, the quantity and quality of the live coal and ash on the grates, the water level, and the steam pressure, should be as nearly as possible the same at the end as at the beginning of the test.

To secure the desired equality of conditions with hand-fired boilers, the following method should be employed:

The furnace being well heated by a preliminary run, burn the fire low, and thoroughly clean it, leaving enough live coal spread evenly over the grate (say from two to four inches)[1] to serve as a foundation for the new fire. Note quickly (1) the thickness of the coal bed as nearly as it can be estimated or measured; also

[1] 1 to 2 inches for small anthracite coals.

(2) the water level,[1] (3) the steam pressure, and (4) the time, and record the latter as the starting time. (5) Fresh coal should then be fired from that weighed for the test, (6) the ashpit should be thoroughly cleaned, and the regular work of the test proceeded with.

Before the end of the test the fire should again be burned low and cleaned in such a manner as to (1) leave the same amount of live coal on the grate as at the start. When this condition is reached, observe quickly (2) the water level,[1] (3) the steam pressure, and (4) the time, and record the latter as the stopping time. If the water level is not the same as at the beginning (5) a correction should be made by computation, rather than by feeding additional water after the final readings are taken. Finally (6) remove the ashes and refuse from the ashpit.

In a plant containing several boilers where it is not practicable to clean them simultaneously, the fires should be cleaned one after the other as rapidly as may be, and each one after cleaning charged with enough coal to maintain a thin fire in good working condition. After the last fire is cleaned and in working condition, burn all the fires low (say 4 to 6 in.), note quickly the thickness of each, also the water levels, steam pressure, and time, which last is taken as the starting time. Likewise when the time arrives for closing the test, the fires should be quickly cleaned one by one, and when this work is completed they should all be burned low the same as at the start, and the various observations made as noted.

In the case of a large boiler having several furnace doors requiring the fire to be cleaned in sections one after the other, the above directions pertaining to starting and stopping in a plant of several boilers may be followed.

To obtain the desired equality of conditions of the fire when a mechanical stoker other than a chain grate is used, the procedure should be modified where practicable as follows:

Regulate the coal feed so as to burn the fire to the low condition required for cleaning. Shut off the coal-feeding mechanism and fill the hoppers level full. Clean the ash or dump plate, note quickly the depth and condition of the coal on the grate, the water level,[1] the steam pressure, and the time, and record the latter as the starting time. Then start the coal-feeding mechanism, clean the ashpit, and proceed with the regular work of the test.

When the time arrives for the close of the test, shut off the coal-feeding mechanism, fill the hoppers and burn the fire to the same low point as at the beginning. When this condition is reached, note the water level, the steam pressure, and the time, and record the latter as the stopping time. Finally clean the ash plate and remove the ashes.

In the case of chain grate stokers, the desired operating conditions should be maintained for half an hour before starting a test and for a like period before its close, the height of the throat plate and the speed of the grate being the same during both of these periods.

The coal should be weighed and delivered to the firemen in portions sufficient for one hour's run, thereby ascertaining the degree of uniformity of firing. An ample supply of coal should be maintained at all times, but the quantity on

[1] Do not blow the water-glass column for at least one hour before these readings are taken. An erroneous indication may otherwise be caused by a change of temperature and density of the water within the column and connecting pipe.

the floor at the end of each hour should be as small as practicable, so that the same may be readily estimated and deducted from the total weight.

The records should be such as to **ascertain also the consumption of feedwater each hour,** and thereby determine the degree of uniformity of evaporation.

Ashes and Refuse. The ashes and refuse withdrawn from the furnace and ashpit during the progress of the test and at its close should be weighed so far as possible in a **dry state.** If wet the amount of moisture should be ascertained and allowed for, a sample being taken and dried for this purpose. This sample may serve also for analysis and the determination of unburned carbon and fusing temperature.

Analyses of Flue Gases. For approximate determinations of the composition of the flue gases, a portable type of apparatus should be employed. If momentary samples are obtained the analyses should be made as frequently as possible, say every 15 to 30 minutes, depending on the skill of the operator, **noting at the time the sample is drawn the furnace and firing conditions.** If the sample drawn is a continuous one, the intervals may be made longer.

Smoke Observations. In tests of bituminous coals requiring a determination of the amount of smoke produced, observations should be made regularly throughout the trial at intervals of five minutes (or if necessary every minute), noting at the same time the furnace and firing conditions.

Calculation of Results. The methods to be followed in expressing and calculating those results which are not self-evident are explained as follows:

(a) *Efficiency.* The "efficiency of boiler, furnace and grate" is the relation between the heat absorbed per pound of coal fired, and the calorific value of one pound of coal.

The "efficiency of boiler and furnace" is the relation between the heat absorbed per pound of combustible burned, and the calorific value of one pound of combustible. This expression of efficiency furnishes a means for comparing one boiler and furnace with another, when the losses of unburned coal due to grates, cleanings, etc., are eliminated.

The " combustible burned" is determined by subtracting from the weight of coal supplied to the boiler, the moisture in the coal, the weight of ash and unburned coal withdrawn from the furnace and ashpit, and the weight of dust, soot, and refuse, if any, withdrawn from the tubes, flues, and combustion chambers, including ash carried away in the gases, if any, determined from the analyses of coal and ash. The " combustible " used for determining the calorific value is the weight of coal less the moisture and ash found **by analysis.**

The " heat absorbed " per pound of coal, or combustible is calculated by multiplying the equivalent evaporation from and at 212 degrees per pound of coal or combustible by 970.4.

(b) *Corrections for Moisture in Steam.* When the percentage is less than 2 per cent it is sufficient merely to deduct the percentage from the weight of water fed. If

the percentage is greater than 2 per cent or if extreme accuracy is required, the factor of correction equals

$$X + P\frac{(q_1 - q_2)}{(H - q_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (85)$$

in which X is the quality of the steam (one minus the decimal representing the percentage of moisture), P the proportion of moisture,[1] q_1 the total heat of water at the temperature of the steam, q_2 the total heat of the feedwater, and H the total heat of saturated steam of the given temperature.

(c) *Correction for live steam, if any, used for aiding Combustion.* If live steam is admitted into the furnace or ashpit for producing blast, injecting fuel, or aiding combustion, it is to be deducted from the total evaporation, and the net evaporation used in the various calculations.

(d) *Equivalent Evaporation.* The equivalent evaporation from and at 212 deg. is obtained by multiplying the weight of water evaporated, corrected for moisture in steam, by the **"factor of evaporation."** The latter equals

$$\frac{H - q_2}{970.4},$$

in which H and q_2 are respectively the total heat of saturated steam and of the feedwater entering the boiler. When the steam is superheated, the total heat of the steam is that of saturated steam plus the product of the number of degrees of superheating by the specific heat of the steam.

Unless otherwise provided, a combined boiler and superheater should be treated as one unit, and the equivalent of the work done by the superheater should be included in the evaporative work of the boiler.

(e) *Heat Balance.* The "heat balance," or approximate distribution of the calorific value of the coal or combustible among the several items of heat utilized and heat lost, should be obtained in cases where the flue gases have been analyzed and a complete analysis made of the coal.

The loss due to moisture in the coal is found by multiplying the difference between the total heat of one pound of superheated steam at the temperature of the escaping gases and the temperature of the air in the boiler room, by the weight of moisture in a pound of coal.

The **loss due to moisture** formed by the burning of hydrogen is obtained by multiplying the total heat of one pound of superheated steam at the temperature of the escaping gases, calculated from the temperature of the air in the boiler room, by the proportion of the hydrogen, determined from the analysis of the coal, and multiplying the result by 9.

The loss due to heat carried away in the dry gases is found by multiplying the weight of gas per pound of coal or combustible by the elevation of temperature of the gases above the temperature of the boiler room, and by the specific heat of the gases (0.24). The weight of gas referred to is obtained by finding the weight of dry gas per pound of carbon burned, using the formula

$$\frac{11\,CO_2 + 8\,O + 7\,(CO + N)}{3\,(CO_2 + CO)}, \text{ (86), page 281,}$$

in which CO_2, CO, O, and N are expressed in percentages by volume, and multiplying this result by the proportion borne by the carbon burned to the whole amount of coal or combustible as determined from the results of the analysis of the coal, ash and refuse.

[1] Proportion of moisture is the ratio of the percentage of moisture in the steam to 100.

The loss due to incomplete combustion of carbon is found by first obtaining the proportion borne by the carbon monoxide in the gases to the sum of the carbon monoxide and carbon dioxide, and then multiplying this proportion by the proportion of carbon in the coal or combustible, and finally multiplying the product by 10,150, which is the number of heat units generated by burning to carbon dioxide one pound of carbon contained in carbon monoxide.

The loss due to combustible matter in the ash and refuse is found by multiplying the proportion that this combustible bears to the whole amount of coal or combustible, by its calorific value per pound. For most purposes it is sufficient to assume the combustible to be 14,600 B.t.u. per pound, the same as that of carbon.

The loss due to moisture in the air is determined by multiplying the weight of such moisture per pound of coal or combustible by the elevation of temperature of the flue gases above the temperature of the boiler room and by 0.47. The weight of moisture is found by multiplying the weight of air per pound of coal or combustible by the moisture in one pound of air determined from readings of the wet- and dry-bulb thermometer.

(*f*) *Total Heat of Combustion of Coal, by Analysis.* The total heat of combustion may be computed from the results of the ultimate analysis by using the formula

$$14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right) + 4000\,S, \quad (69), \text{ page } 227,$$

in which C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen and sulphur, respectively.

(*g*) *Air for Combustion.* The quantity of air used may be calculated by the formulæ:

$$\text{Pounds of air per pound of carbon} = \frac{3.032\,N}{CO_2 + CO},$$

in which N, CO_2 and CO are the percentages of dry gas obtained by analysis, and

Lbs. of air per lb. of coal = lbs. air per lb. C $\times$ per cent C in the coal.

The ratio of the air supply to that theoretically required for complete combustion is $\dfrac{N}{N - 3.782\,O}$. Compare with formulas on page 249.

Tests with Oil and Gas Fuels. Tests of boilers using oil or gas for fuel should accord with the rules here given, excepting as they are varied to conform to the particular characteristics of the fuel. The duration in such cases may be reduced, and the "flying" method of starting and stopping employed.

The table of data and results should contain items stating character of furnace and burner, quality and composition of oil or gas, temperature of oil, pressure of steam used for vaporizing and quantity of steam used for both vaporizing and for heating.

TABLE 1. DATA AND RESULTS OF EVAPORATIVE TEST — SHORT FORM CODE OF 1912

(1) Test of........................boiler located at........................
to determine....................conducted by......................
(2) Kind of furnace..
(3) Grate surface..sq. ft.

(4) Water-heating surface[1] .... sq. ft.
(5) Superheating surface[1] .... sq. ft.
(6) Date ....
(7) Duration .... hrs.
(8) Kind and size of coal ....

AVERAGE PRESSURES, TEMPERATURES, ETC.

(9) Steam pressure by gage .... lbs. per sq. in.
(9a) Absolute steam pressure .... lbs. per sq. in.
(10) Temperature of feedwater entering boiler .... deg. F.
(11) Temperature of escaping gases leaving boiler .... deg. F.
(12) Force of draft between damper and boiler .... ins. water
(13) Percentage of moisture in steam, or number deg. of superheating (per cent or deg. F.)

TOTAL QUANTITIES

(14) Weight of coal as fired[2] .... lbs.
(15) Percentage of moisture in coal .... per cent
(16) Total weight of dry coal consumed .... lbs.
(17) Total ash and refuse .... lbs.
(18) Percentage of ash and refuse in dry coal .... per cent
(19) Total weight of water fed to the boiler[3] .... lbs.
(20) Total water evaporated, corrected for moisture in steam .... lbs.
(21) Total equivalent evaporation from and at 212 deg. F. .... lbs.

HOURLY QUANTITIES AND RATES

(22) Dry coal consumed per hour .... lbs.
(23) Dry coal per sq. ft. of grate surface per hour .... lbs.
(24) Water evaporated per hour corrected for quality of steam .... lbs.
(25) Equivalent evaporation per hour from and at 212 deg. F .... lbs.
(26) Equivalent evaporation per hour from and at 212 deg. F. per sq. ft. of water-heating surface .... lbs.

CAPACITY AND ECONOMY RESULTS

(27) Evaporation per hour from and at 212 deg. F. (same as Line 25) .... lbs.
(28) Boiler horse power developed (Item 27 ÷ 34½) .... bl.h.p.
(29) Rated capacity, in evaporation from and at 212 deg. F. per hour .... lbs.
(30) Rated boiler horse power .... bl.h.p.
(31) Percentage of rated capacity developed .... per cent
(32) Water fed per lb. of coal fired (Item 19 ÷ Item 14) .... lbs.
(33) Water evaporated per lb. of dry coal (Item 20 ÷ Item 16) .... lbs.
(34) Equivalent evaporation from and at 212 deg. F. per lb. of dry coal (Item 21 ÷ Item 16) .... lbs.
(35) Equivalent evaporation from and at 212 deg. F. per lb. of combustible [Item 21 ÷ (Item 16 − Item 17)] .... lbs.

[1] See page 258 for definition of heating surfaces.

[2] The term "as fired" means actual condition including moisture, corrected for estimated difference in weight of coal on the grate at beginning and end of test.

[3] Corrected for inequality of water level and steam pressure at beginning and end of test.

EFFICIENCY

(36) Calorific value of 1 lb. of dry coal.................................B.t.u.
(37) Calorific value of 1 lb. of combustible.................................B.t.u.
(38) Efficiency of boiler, furnace and grate $\left[100 \times \frac{\text{Item 34} \times 970.4}{\text{Item 36}}\right]$.........per cent
(39) Efficiency of boiler and furnace $\left[100 \times \frac{\text{Item 35} \times 970.4}{\text{Item 37}}\right]$..............per cent

COST OF EVAPORATION

(40) Cost of coal per ton of .... lbs. delivered in boiler room................dollars
(41) Cost of coal required for evaporating 1000 lbs. of water from and at 212 deg., dollars

TABLE 2. DATA AND RESULTS OF EVAPORATIVE TEST — COMPLETE FORM, CODE OF 1912

(1) Test of........................boiler located at.........................
to determine......................conducted by.....................

DIMENSIONS, PROPORTIONS, ETC.

(2) Number and kind of boilers...
(3) Kind of furnace..
(4) Grate surface........width........length........area................sq. ft.
(5) Approximate width of air spaces in grate................................ins.
(6) Proportion of air space to whole grate surface......................per cent
(7) Water-heating surface...sq. ft.
(8) Superheating surface..sq. ft.
(9) Ratio of water-heating surface to grate surface..........................to 1
(10) Ratio of minimum draft area to grate surface....................1 to.......
(11) Date...
(12) Duration...hrs.
(13) Kind of coal..
(14) Size of coal..

AVERAGE PRESSURES, TEMPERATURES, QUALITY OF STEAM, ETC.

(15) Steam pressure by gage...................................lbs. per sq. in.
(16) Barometric pressure................ins. mercury =..........lbs. per sq. in.
(16a) Absolute steam pressure..................................lbs. per sq. in.
(17) Force of draft at dampers of individual boilers.....................ins. water
(18) Force of draft in main flue near boilers.........................ins. water
(19) Force of draft in main flue between economizer and chimney........ins. water
(20) Force of draft in furnaces.....................................ins. water
(21) Force of blast in ashpits......................................ins. water
(22) State of weather...
(23) Temperature of external air.....................................deg. F.
(24) Temperature of fireroom..deg. F.
(25) Temperature of steam...deg. F.
(26) Normal temperature of saturated steam............................deg. F.
(27) Temperature of feedwater entering flue heater or economizer..........deg. F.
(28) Temperature of feedwater leaving heater or economizer and entering boilers...deg. F.

(29) Temperature of gases leaving boilers. . . . . deg. F.
(30) Temperature of gases leaving economizer. . . . . deg. F.
(31) Percentage of moisture in steam. . . . . per cent
(32) Number of degrees of superheating. . . . . deg. F.
(33) Quality of steam (dry steam = unity). . . . .

Total Quantities

(34) Weight of coal as fired[1]. . . . . lbs.
(35) Percentage of moisture in coal. . . . . per cent
(36) Total weight of dry coal consumed. . . . . lbs.
(37) Total ash and refuse. . . . . lbs.
(38) Total combustible consumed (Line 36 − Line 37). lbs.
(39) Percentage of ash and refuse in dry coal. . . . . per cent
(40) Total weight of water fed to boiler[2]. . . . . lbs.
(41) Total water evaporated corrected for moisture in steam. . . . . lbs.
(42) Factor of evaporation, based on temperature of water entering boilers. . . . .
(43) Total equivalent evaporation from and at 212 deg. F. lbs.

Hourly Quantities and Rates

(44) Dry coal consumed per hour. . . . . lbs.
(45) Combustible consumed per hour. . . . . lbs.
(46) Dry coal per sq. ft. of grate surface per hour. . . . . lbs.
(47) Water evaporated per hour, corrected for quality of steam. . . . . lbs.
(48) Equivalent evaporation per hour from and at 212 deg.[3] F. . . . . lbs.
(49) Equivalent evaporation per hour and at 212 deg. F. per sq. ft. of water-heating surface[3]. . . . . lbs.

Proximate Analysis of Coal

(50) Fixed carbon. . . . . per cent
(51) Volatile matter. . . . . per cent
(52) Moisture. . . . . per cent
(53) Ash. . . . . per cent

100 per cent

(54) Sulphur, separately determined. . . . . per cent

Ultimate Analysis of Dry Coal

(55) Carbon (C). . . . . per cent
(56) Hydrogen (H). . . . . per cent
(57) Oxygen (O). . . . . per cent
(58) Nitrogen (N). . . . . per cent
(59) Sulphur (S). . . . . per cent
(60) Ash. . . . . per cent

100 per cent

(61) Moisture in sample of coal as received. . . . . per cent

[1] The term "as fired" means actual condition including moisture, corrected for difference in weight of coal on grate at beginning and end of test.

[2] Corrected for inequality of water level and steam pressure at beginning and end of test.

[3] The symbol U.E. meaning "units of evaporation" (see page 268) may be substituted for the expression, equivalent water evaporated into dry steam from and at 212 deg. Fahrenheit.

Analysis of Ash and Refuse

(62) Carbon...per cent
(63) Earthy matter..per cent
(64) Temperature of fusion of ash.................................deg. F.

Calorific Value

(65) Calorific value of 1 lb. of dry coal by calorimeter........B.t.u.
(66) Calorific value of 1 lb. of combustible by calorimeter.....B.t.u.
(67) Calorific value of 1 lb. of dry coal by analysis...........B.t.u.
(68) Calorific value of 1 lb. of combustible by analysis........B.t.u.

Capacity, Economy Results and Efficiency

(69) Evaporation per hour from and at 212 deg. F. (same as Line 48)...........lbs.
(70) Boiler horse power developed (Line 69 ÷ 34½) ....bl.h.p.
(71) Rated capacity per hour, from and at 212 deg. F..........lbs.
(72) Rated boiler horse power...................................bl.h.p.
(73) Percentage of rated capacity developed.....................per cent
(74) Water fed per lb. of coal (Item 40 ÷ Item 34)..............lbs.
(75) Water evaporated per lb. of dry coal (Item 41 ÷ Item 36) ....lbs.
(76) Equivalent evaporation from and at 212 deg. F. per lb. of coal fired (Item 43 ÷ Item 34)..........lbs.
(77) Equivalent evaporation from and at 212 deg. F. per lb. of dry coal (Item 43 ÷ Item 36)..........lbs.
(78) Equivalent evaporation from and at 212 deg. F. per lb. of combustible (Item 43 ÷ Item 38)..........lbs.
(79) Efficiency of boiler, furnace, and grate $\left[100 \times \frac{\text{Item 77} \times 970.4}{\text{Item 65}}\right]$........per cent
(80) Efficiency of boiler and furnace $\left[100 \times \frac{\text{Item 78} \times 970.4}{\text{Item 66}}\right]$..............per cent

Cost of Evaporation

(81) Cost of coal per ton of .... lbs. delivered in boiler room..............dollars
(82) Cost of coal required for evaporating 1000 lbs. of water under observed conditions..........dollars
(83) Cost of coal required for evaporating 1000 lbs. of water from and at 212 deg. F..........dollars

Smoke Data

(84) Percentage of smoke as observed...........................per cent
(85) Weight of soot per hour obtained from smoke meter..........................

Methods of Firing

(86) Kind of firing, whether spreading, alternate, or coking.......................
(87) Average thickness of fire..................................ins.
(88) Average intervals between firings for each furnace during time when fires are in normal condition..........min.
(89) Average interval between times of leveling or breaking up................min.

Analysis of Dry Gases by Volume

(90) Carbon dioxide (CO_2)....................................per cent
(91) Oxygen (O)....................................per cent
(92) Carbon monoxide (CO)....................................per cent
(93) Hydrogen and hydrocarbons....................................per cent
(94) Nitrogen, by difference (N)....................................per cent
100 per cent

HEAT BALANCE BASED ON DRY COAL AND COMBUSTIBLE

	Dry Coal as Fired.		Combustible Burned.	
	B.t.u.	Per Cent.	B.t.u.	Per Cent.
(95) Heat absorbed by the boiler (Line 77 or 78 × 970.4)........				
(96) Loss due to evaporation of moisture in coal (page 275).......				
(97) Loss due to heat carried away by steam formed by the burning of hydrogen (page 275)........				
(98) Loss due to heat carried away in the dry flue gases.........				
(99) Loss due to carbon monoxide........				
(100) Loss due to combustible in ash and refuse........				
(101) Loss due to heating moisture in air........				
(102) Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for........				
(103) Total calorific value of 1 lb. of dry coal or combustible (Lines 65 and 66)........		100		100

COMPUTATION OF THE WEIGHT OF THE CHIMNEY GASES FROM THE ANALYSIS BY VOLUME OF THE DRY GAS

Two methods of calculating from the analysis by volume of the dry chimney gases the number of pounds of dry chimney gases per pound of carbon, or the weight of air supplied per pound of carbon, have been given by different writers. These may be expressed in the shape of formulas as follows:

$$\text{(A) Pounds dry gas per pound C} = \frac{11\,CO_2 + 8\,O + 7\,(CO + N)}{3\,(CO_2 + CO)} \qquad (86)$$

$$\text{(B) Pounds air per pound C} = 5.8\,\frac{2(CO_2 + O) + CO}{CO_2 + CO} \quad . \quad . \quad . \quad . \qquad (87)$$

in which CO_2, O_2, CO and N are percentages by volume of the gases.

Formula A may be derived from the method of computation given in Mr. R. S. Hale's paper on "Flue-Gas Analyses," *Transactions American Society of Mechanical Engineers*, vol. 18, page 902, and formula B from the method given in Peabody's and Miller's "*Treatise on Steam Boilers.*" Both are based on the principle that the density, relatively to hydrogen, of an elementary gas (O and N) is proportional to its atomic weight, and that of a compound gas (CO and CO_2) to one-half its molecular weight. Both formulas are very nearly accurate when pure carbon is the fuel burned, but formula B is inaccurate when the fuel contains hydrogen, for the reason that the portion of the oxygen of the air supply which is re-

quired to burn the hydrogen is contained in the chimney gas as water vapor and does not appear in the analysis of the dry gas.

The following calculations of a supposed case of combustion of hydrogenous fuel illustrates the accuracy of formula A and the inaccuracy of formula B. Assume that the coal has the following analysis: C, 66.50; H, 4.55; O, 8.40; N, 1.00; water, 10.00; ash and sulphur, 9.55—total, 100. Assume that one-tenth of the C is burned to CO, and nine-tenths to CO_2; that the air supply is 20 per cent in excess of that required for this combustion; that the air contains 1 per cent by weight of moisture; and that the S in the coal may be considered as part of the ash. We then have the following summary of results of the combustion of 100 pounds of coal:

	O from Air.	N = O × [illegible].	Total Air.	CO_2.	CO.	H_2O.
59.85 lbs. C to $CO_2 \times 2\frac{2}{3}$	159.60	534.31	693.91	219.45	......	......
6.65 " C to $CO \times 1\frac{1}{3}$	8.87	29.70	38.57	....	15.52	......
3.50 " H to $H_2O \times 8$	28.00	93.74	121.74	....	......	31.50
	196.47	657.75	854.22	....	......	......
1.05 " H to H_2O						
8.40 " H to H_2O	....	....	....	....	......	9.45
10.00 " Water	....	....	....	....	......	10.00
1.00 " N	....	1.00	....	....	......	......
9.55 " Ash and S	....	....	....	....	......	......
100.00						
Excess of air 20 per cent.	39.29	131.55	170.84	....	......	......
	....	....	1025.06	....	......	......
Moisture in air 1 per cent.	....	....	....	....	......	10.25
Total wt. gases, 1125.76 lbs. =	39.29	790.30	....	219.45	15.52	61.20
Total dry gases, 1064.56 lbs.						
Per Cent	O	N		CO_2	CO	
Total dry gases, by weight,	3.69	74.24	....	20.61	1.546	......
Total dry gases, by volume,	3.508	80.656	....	14.252	1.584	......

Total gases 1125.76 + ash and S 9.55 = 1135.31 lbs. total products.
Total air 1025.06 + moisture in air 10.25 + coal 100 = 1135.31 lbs.
Dry gas per lb. coal 10.6456; per lb. carbon = 10.6456 ÷ .665 = 16.008 lbs.
Dry air per lb. coal 10.2506; per lb. carbon = 10.2506 ÷ .665 = 15.414 lbs.

Computation of the weight of dry gas and of air per lb. carbon.

Formula A:

$$\text{Dry gas per lb. C} = \frac{14.252 \times 11 + 3.508 \times 8 + 82.240 \times 7}{3\,(14.252 + 1.584)} = 16.008 \text{ pounds.}$$

Formula B:

$$\text{Air per pound C} = 5.8\,\frac{2\,(14.252 + 3.508) + 1584}{14.252 + 1.584} = 13.589 \text{ pounds.}$$

The error in the last result is 15.414 − 13.589 = 1.825 pounds.

Professor D. S. Jacobus gives another formula for the air per pound of carbon, in which the error of formula 87 is almost entirely avoided.

It is

Formula C:

$$\text{Air per pound C} = \frac{7\,N}{3\,(CO_2 + CO)} \div 0.77, \text{ or } \frac{N}{0.33\,(CO_2 + CO)}, \tag{88}$$

in which N, CO_2, and CO are the percentages by volume of these gases. Making the computation from the data of the above analysis, we have:

$$\text{Air per pound C} = \frac{80.656}{0.33\,(14.252 + 1.584)} = 15.434 \text{ pounds},$$

the true value being 15.414 pounds.

CHAPTER XII

STEAM ENGINE TESTING

Most important of the tests made of nearly all classes of machinery is that for **mechanical effic ency**; meaning the comparison of the useful work performed with the amount of work theoretically possible to obtain with a perfect machine. In other words, in an engine the mechanical efficiency, E_m, is the ratio of the brake horse power to the indicated horse power, or

$$E_m = \frac{\text{b.h.p.}}{\text{i.h.p.}} \quad \ldots \ldots \ldots \quad (89)$$

STEAM ENGINE TESTING

TESTS FOR MECHANICAL EFFICIENCY AND FRICTION

Date............................Test made by..........................

Description of engine tested..

....

Tare of Brake................lbs. Length of brake arm..........feet..........

Engine and Brake Constants (see pages 143 and 148)........................

No. of Reading.	Time.	Weight on Brake, lbs.		R.P.M.	Areas of Indicator, Cards, sq. ins.		Indicated Horse Power.			Brake Horse Power.	Friction Horse Power.	Mech. Effic. %
		Gross.	Net.		Head End.	Crank End.	Head End.	Crank End.	Total.			

The difference between the indicated horse power and the brake horse power is called the **friction horse power.** In many cases with very large engines, it is not readily possible to obtain the brake horse power directly, and in such cases it is customary to obtain approximately the horse power lost in friction from a so-called "friction indicator diagram," obtained from the areas of indicator diagrams when the only work done is that required to overcome its own friction, or in common parlance, when the engine is "running light." The brake horse power is then taken to be the difference between the indicated horse power

and the friction horse power. Such a determination of friction horse power and of mechanical efficiency by calculation cannot be considered very accurate, because the friction of an engine increases slightly with increasing loads.

Observed and calculated data of mechanical efficiency may be tabulated as shown in table on page 284.

Valve Setting (Slide Valve Engines). In order that steam may be used economically in an engine, it is necessary that the valve be set carefully and accurately, so that when an indicator card is taken the diagram obtained will be as nearly as possible like the ideal Adjustment of a slide valve on an engine is accomplished in two different ways, with different effects:

(1) By moving the valve on its stem;

(2) By adjusting the eccentric.

Typical slide valves are shown in Figs. 297 and 298.

Fig. 297. — Ordinary D-slide Valve in Mid-position.

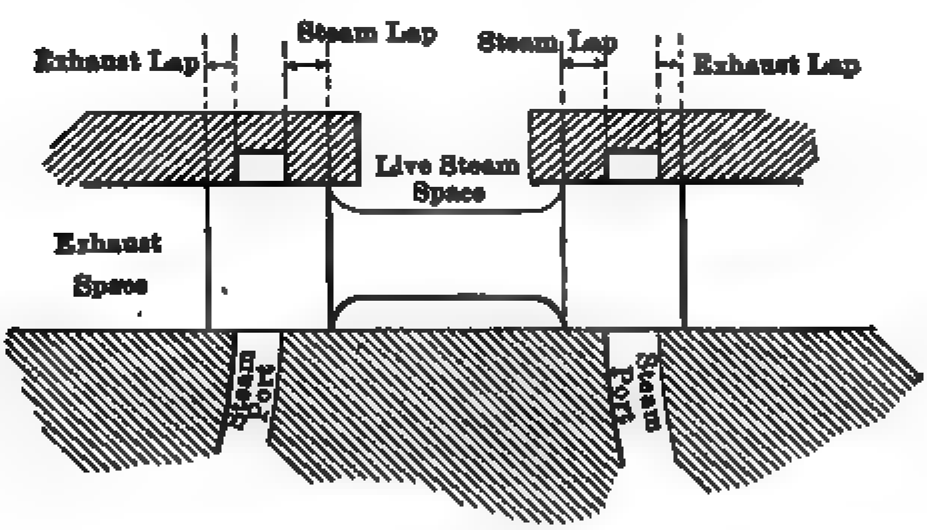


Fig. 298. — Piston Type of Slide Valve in Mid-position.

To Set the Valve for Equal Leads. The first step in setting a valve is to place the engine on dead-center and adjust the angle between the crank and eccentric so that the valve opens the port leading to the cylinder a slight amount. The width of the opening should be measured and recorded as a preliminary value of the lead on that end,[1] — suppose

[1] It is assumed of course that corresponding dimensions of the ports are the same at the two ends of the valve seat.

for example it is $\frac{1}{8}$ inch. Then the engine should be placed on the opposite dead-center and the port opening on that end measured and recorded as the preliminary value of the lead on that end, suppose it is $\frac{1}{16}$ inch. There is then a difference in lead on the two ends of $\frac{1}{16}$ inch. The valve must be moved on its stem a distance equal to half the difference, or $\frac{1}{32}$ inch. This movement of the valve will be in a direction **away from** the port having the **smaller** opening.

By the method described the two leads of the valve will be made the same; in other words, the distance the valve uncovers the steam ports when the engine is on the dead-center will be the same at both ends of the cylinder. But while the leads are equal they are not necessarily the required amount and it remains to set the eccentric to give the leads desired. Place the engine once more accurately on dead-center[1] and after loosening the eccentric move it on the shaft so as to change the lead to the amount desired. As a final check, after securing the eccentric, the engine can be placed on the other dead-center to see that the lead is correct.[2]

To set the valve for equal cut-offs, the valve is first set on its stem so that the travel will be the same on both sides of the mid-position as explained at the bottom of this page to make its movement symmetrical with the ports. Now an adjustment of the eccentric must be made so that steam will be cut off at each end of the cylinder when the piston has moved the same distances or the same per cent of the stroke from the two ends. To perform this adjustment mark on the cross-head as accurately as possible the limits of the stroke, and set the cross-head at the per cent of the stroke for cut-off appearing to be most suitable for the conditions of load. Move the eccentric on the shaft in the direction

[1] An engine can be put on dead-center quite accurately by the "method of trammels." When the engine is just a little off the center to be determined, make small scratch-marks opposite each other both on the cross-head and on one of the guides. Now set a pair of dividers or trammels with one end resting on the bedplate of the engine, its foundation, or some convenient stationary object near the fly-wheel, and with the other end mark a point on the fly-wheel. The engine should then be moved over or beyond the dead-center until the marks made on the cross-head and on the guide come together again. With the dividers set with the points the same distance apart as before again put a mark on the fly-wheel. Then if the engine is turned back so that the end of the dividers used to mark on the fly-wheel is at a point half way between the two marks, it will be set quite accurately on the dead-center required. In all these adjustments care must be taken to turn the engine each time in the same direction with respect to the dead-center so that the lost motion or back lash is taken up in the same direction. The engine must be placed accurately on center because when the crank is near the dead-center the eccentric is in such a position that a slight movement of the shaft causes considerable movement of the valve.

[2] This method applies only explicitly to a valve like the one in **Fig. 297**, which takes steam on the "outside." When the valve takes steam on the inside (**Fig. 298**) the eccentric must be moved in the opposite direction.

in which the engine is to run until it can be seen that the valve would be just closing the steam port at the end of the cylinder from which the piston is moving. Fasten the eccentric securely in this position and turn the engine over to observe whether the valve will be just closing the other steam port when the piston has moved the same distance, measured on the cross-head, from the other end of the cylinder. If the setting is not correct, the error should be halved, correcting for one half the error by moving the valve on the stem and for the other half by moving the eccentric on the shaft. This operation, which is a "cut and try" process, must be repeated until the required setting is secured.

Methods similar to those described are preferred usually for the accurate setting of the slide valves of slow- and medium-speed engines. High-speed engines as well as slow-speed engines of the Corliss type have their valves set usually on the basis of the information secured

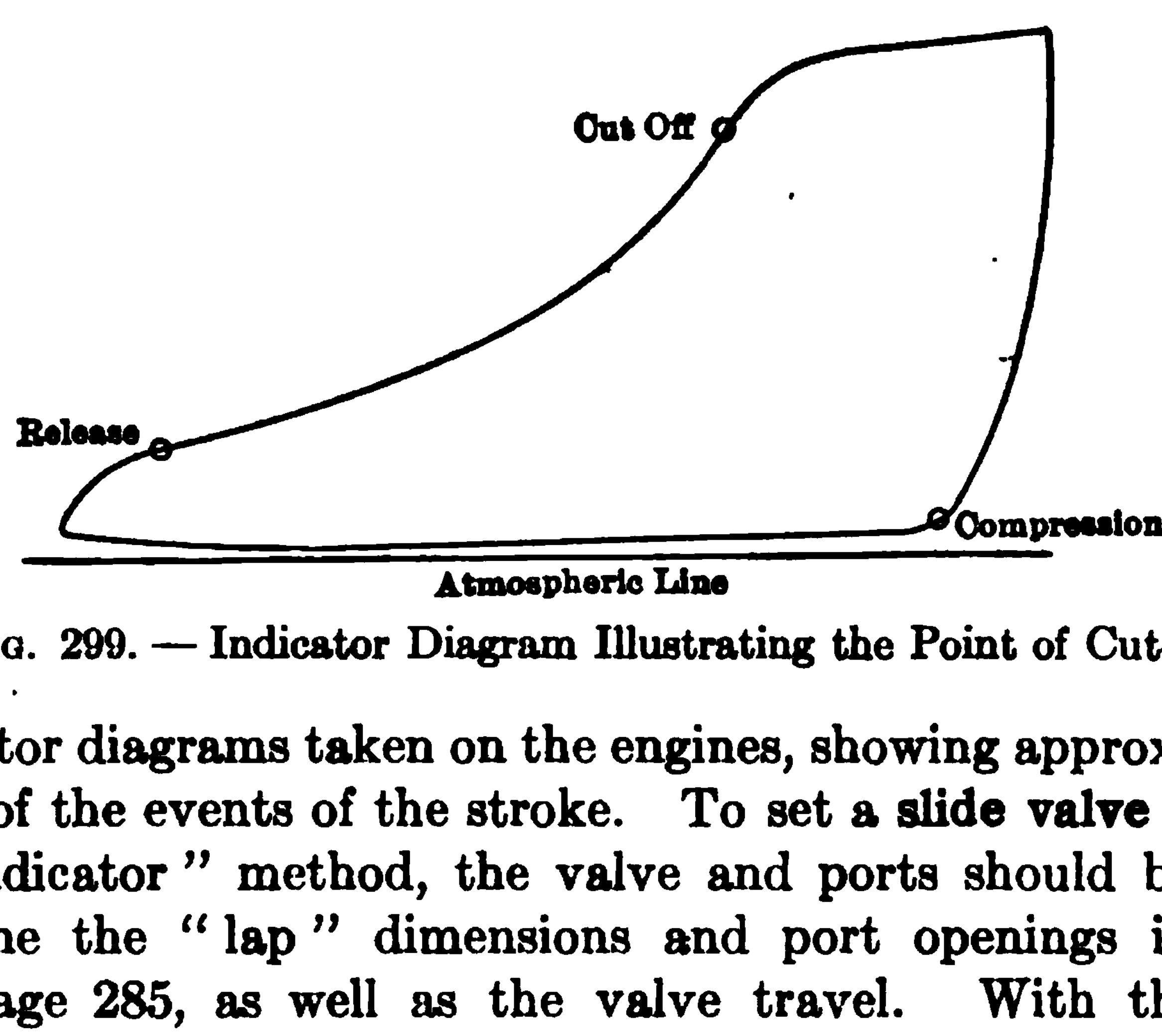


FIG. 299. — Indicator Diagram Illustrating the Point of Cut-off.

from indicator diagrams taken on the engines, showing approximately the "timing" of the events of the stroke. To set a **slide valve** successfully by the "indicator" method, the valve and ports should be measured to determine the "lap" dimensions and port openings indicated in **Fig. 297** page 285, as well as the valve travel. With these data a **Zeuner[1] valve diagram** should be constructed, showing a good steam distribution for assumed lead or cut-off. Then construct the theoretical indicator card from the Zeuner diagram and adjust the setting of the valve on the stem and the eccentric on the shaft until a close approximation to the theoretical card is obtained. In this adjustment the first thing to be done is to equalize the travel of the valve by locating it on its stem so that the travel will be the same on both sides of its mid-position.

Use a spring in the indicator light enough to give a diagram about 1½

[1] It is beyond the scope of this book to take up a discussion of valve diagrams. The theory and construction of the Zeuner diagram are given in nearly all books on the steam engine. Bilgram valve diagrams, although excellent for designers, are not as good as the Zeuner diagram for valve setting requirements.

inches high so that events of the stroke, — admission, cut-off, release, and compression, will be shown as clearly as possible. Sometimes it is difficult to determine these events on a diagram on account of the curves gradually running into each other without the point separating the different curves being clearly defined. A good method for such cases is to produce along their regular trend both of the curves of which the intersection is required and take for the intersection the point where these curves cross each other. The method is illustrated on an indicator card in **Fig. 299** showing the point of cut-off.

In a slide-valve engine it is not possible to set the valve to secure at the same time equal cut-offs and equal leads.

Ideal and imperfect indicator diagrams taken from slide-valve and Corliss engines are shown in **Fig. 300.** A little study of such diagrams may help to solve many difficulties in valve setting.

Setting Corliss Valves. A brief description[1] of the essential parts of the valve gear of a Corliss engine will assist in obtaining a clearer conception of the subject. In **Figs. 301** and **302** similar letters of reference indicate the same parts of the mechanism.

Fig. 301 shows all the essential parts of the valve gear. The steam valves work in the chambers **S, S** and the exhaust valves work in the chambers **E, E.** The double-armed levers **AC, AC** work loosely on the hubs of the valve-stem brackets and the lever arms **B, B;** the former are connected to the wrist plate **W** by the rods **M, M;** the levers **B, B** are keyed to the valve stems **V, V,** and are also connected by the rods **O, O** to the dashpots **D, D.** The double-armed levers carry at their outer ends **C, C** hardened steel catch plates, which engage with arms **B, B,** making the two arms **B** and **C** work in unison until steam is to be cut off; at this point another set of levers **H, H,** connected by the cam rods **G, G** to the governor, come into play, causing the catch plates to release the arms **B, B,** the outer ends of which are then pulled downward by the weight of the dashpot plunger, causing the steam valves to rotate on their axes and thus cut off steam. These are the essential features of the Corliss gear, although the design of the mechanism is greatly modified by different builders.

The exhaust valve arms **F** are connected to the wrist plate by the rods **N, N,** and it is seen that all the valves receive their motion from the wrist plate; the latter receives its motion from the hook rod **I;** this rod is generally attached to a rocker arm, not shown; to this arm the eccentric rod is also attached. The carrier arm is usually placed about midway between the wrist plate and eccentric, and in the center of its travel stands in a vertical position.

[1] This description is mainly from *American Machinist*, vol. 18, page 391. For clearness the article is considered unusually good.

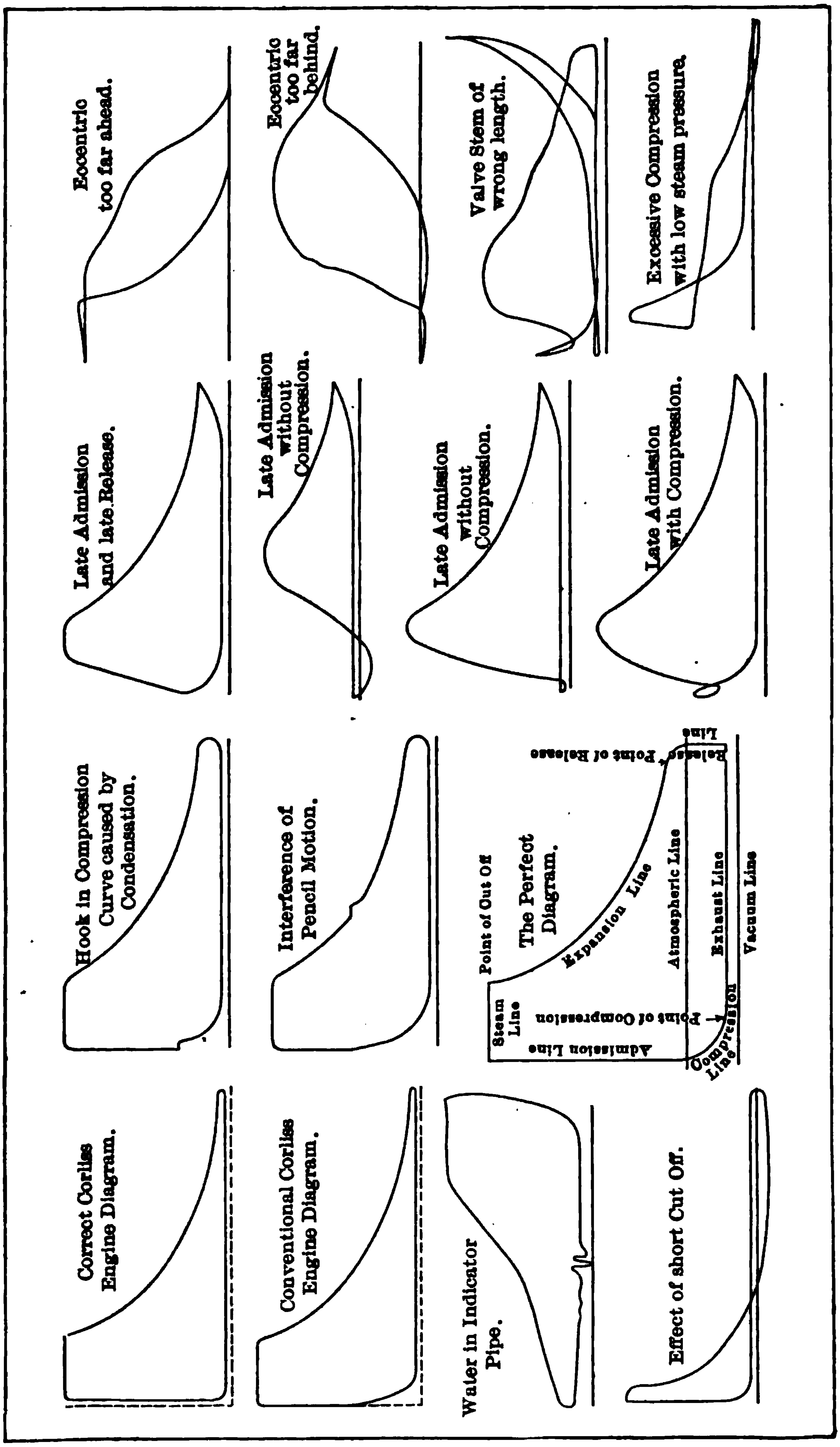


FIG. 300. —Samples of Ideal and Imperfect Indicator Diagrams. (From *Power*.)

The setting of the valves is not a difficult matter when, on the wrist plate, its support, valves and cylinder, the customary marks have been placed for finding the relative positions of wrist plate and valves.

Now referring to **Fig. 302**, when the back bonnets of the valve chambers have been taken off, there will be found a mark or line **a** on the end of each valve **s, s**, coinciding with the working or opening edges of each valve; another line **b** will be found on each face of the steam valve chamber coinciding with the working edge of the valve, and the line **h**, on the face of each exhaust valve chamber, coincides with the working edge of the

FIG. 301. — Corliss Valve Gear.

exhaust port. On the hub of the wrist plate will be found a line **d**, coinciding with the center line **d, k**; lastly, there are three lines **f, c, f**, on the hub of the wrist-plate support, placed in such a way that when the line **d** coincides with the line **c**, the wrist plate will stand exactly in the center of its motion, and when the line **d** coincides with either of the lines **f, f**, the wrist plate will be at one of the extreme ends **u** or **v** of its travel. It should be noticed that since the lines **f, e, f** are drawn on the periphery of the hub of the wrist-plate support, and the line **d** is drawn on the periphery of the wrist-plate hub, these lines cannot stand in a vertical line, as shown. This way of showing them has been adopted simply for the purpose of making the matter plain.

In setting the valves the first step will be to set the wrist plate in its

central position, so that lines **c** and **d** will coincide, and fasten the wrist plate in this position by placing a piece of paper between it and the washer **L** on its supporting pin. Now set the steam valves so that they will have a slight amount of lap, that is to say, the lines **a, a** must have moved a little beyond the lines **b, b**; the amount of this lap depends much on individual preference and experience; it ranges from $\frac{1}{16}$ to $\frac{1}{4}$ inch for small engines, and from $\frac{1}{4}$ to $\frac{9}{16}$ inch for comparatively large engines. This lap is obtained by lengthening or shortening the rods **M, M** by means of the adjusting nuts.

Now by lengthening or shortening the rods **N, N** and by moving the adjusting nuts, place the exhaust valves **e, e,** in a position so that the working edges will just open the exhaust ports, or, in other words, place the lines **g** and **h** nearly in line with each other. Some engineers prefer a slight amount of lap, others prefer a slight opening of the exhaust ports

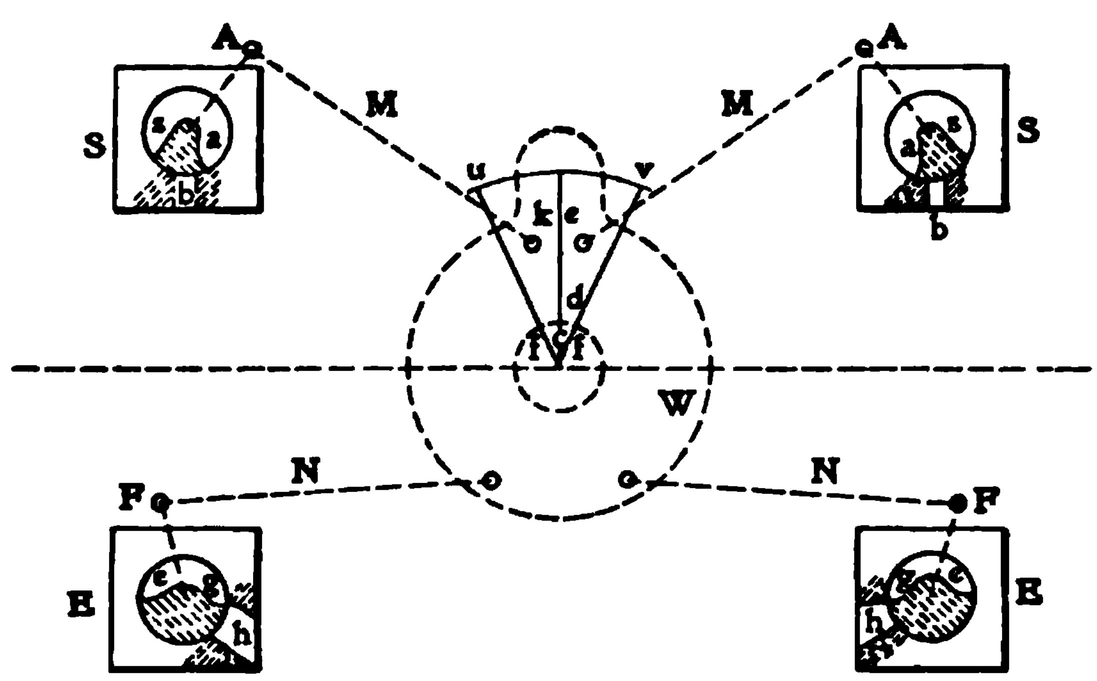


FIG. 302. — Diagram of a Corliss Valve Mechanism.

when the valves are in this position; under these conditions the lines **g** and **h** cannot be in line, but will stand apart, as indicated in the diagram. The distance between these lines will, of course, be equal to the desired amount of opening. For small engines it is about $\frac{1}{16}$ inch, and for larger engines may be increased to $\frac{3}{16}$ inch, but in any case the amount of this opening should be less than the lap of the steam valves, otherwise there will be danger of steam blowing through without doing work.

The paper between the wrist plate and the washer on the supporting pin should now be taken out, so that the wrist plate connected to the valves can be swung on its pin.

The next step will be to give some attention to the rocker arm. Set this arm in a vertical position by means of a plumb-line, and connect the eccentric rod to it, then turn the eccentric around on the shaft, and see that the extreme points of travel are at equal distances from the plumb-line. To secure this a little adjustment in the stub end of the eccentric

rod may be necessary. Now connect the hook rod **I** to its pin on the wrist plate, and again turn the eccentric around on the shaft, and thus determine the extreme points of travel of the wrist plate. If all parts have been correctly adjusted, the line **d** will coincide with the lines **f, f** at the extreme points of travel; if this is not the case, the hook rod will have to be adjusted at its stub end so as to obtain the desired eqaalized motion of the wrist plate.

The next step will be to set the valves correctly with respect to the position of the crank. To do so the lengths of the rods **M, M, N** and **N** must not be changed, but the following procedure should be followed: Place the crank on one of its dead-centers, and turn the eccentric loosely on the shaft in the direction in which the engine is to run, until the steam valve nearest to the piston shows an opening or lead of $\frac{1}{32}$ to $\frac{1}{8}$ inch, according to size of engine, the smaller lead, of course, being adopted for small engines. After the proper lead has been given to this valve, secure the eccentric, and turn the shaft with eccentric in the same direction in which the engine is to run until the crank is on the opposite dead-center, and notice if the opening or lead at this end of the cylinder is the same as on the other steam valve; if not, shorten or lengthen slightly, as may be necessary, the connection between wrist plate and eccentric. Much adjustment in the length of these connections is not permissible without resetting the valves with reference to the wrist plate.

The only thing which now remains to be done is to adjust the cam rods, **G, G**. To do this, secure the governor balls in their highest position, and disconnect the hook rod from wrist pin. Lengthen or shorten the cam rods **G, G,** so as to bring the detachment apparatus into action, swing the wrist plate back and forward and make such adjustments in the rods **G, G,** as to permit the steam valves to be released when the steam port has been opened about $\frac{1}{8}$ inch. This adjustment is for the purpose of keeping the engine under the control of the governor, in case, for some reason or another, the load on the engine is suddenly thrown off. After this adjustment the governor balls should be placed in their lowest position, in which the releasing gear should not detach the steam valves, but allow the steam to follow nearly full stroke. Sometimes the releasing gear is constructed in such a manner as to close the steam valves automatically, in case the belt leading to the governor should be broken, or the load on the engine suddenly thrown off. In cases of this kind the governor balls need not be placed in their highest position, but should be placed in their lowest position, and the wrist plate moved to either end of its extreme travel. The steam port opposite this end of travel of wrist plate will then be wide open. Now adjust the corresponding cam rod so that the releasing gear is nearly on the point of releasing the valve; then move the wrist plate to other end of its extreme travel, and adjust the other cam rod in

the same manner. To prove the correctness of the cut-off adjustment, raise the governor balls to about a position where they would be when at work, or to a medium height, and block them there; then, with the connection made between the eccentric and the wrist plate, turn the engine shaft slowly in the direction in which it is to run, and when the valve is released measure upon the slide the distance through which the cross-head has moved from its extreme position. Continue to turn the shaft in the same direction, and when the other valve is released, measure the distance through which the cross-head has moved from its extreme position, and if the cut-off is equalized, these two distances will be equal to each other. If they are not, adjust the length of the cam rods until the points of cut-off are at equal distances from the beginning of the stroke. Replace the back bonnets and see that all connections have been properly made, which will complete the setting of the valves. Wherever convenient, it is desirable that an indicator be applied to the engine when at work, and the setting of the valves tested. If necessary, they should be readjusted for the best possible condition for economical work.

Clearance Determination of an Engine. This test is made usually to determine the clearance volume of a steam or a gas engine. It is sometimes important to know the clearance volume of an engine, as it materially affects the expansion curve of the engine. If it is too large it causes an excessive loss in the engine. The clearance volume is also necessary if a theoretical expansion curve is to be constructed.

The engine is first set on dead-center with the piston at the head end of the cylinder. This is done by the "method of trammels" (see footnote, page 286). Then the steam chest cover and valve are to be removed and a rubber gasket under a block of wood is placed over both steam-port openings in the valve seat and bolted on. Usually candle wicking must be packed around the piston to stop excessive leakage. For this purpose the cylinder head must be removed and again replaced.

Two vessels filled with clean water should be provided and weighed. The clearance space is to be filled from one vessel, the time required being taken. As soon as the space is filled the first vessel is removed and the space is kept filled with water from the other vessel for five minutes. The vessels are then again weighed and the water used from each of them determined. The average rate of leakage while filling the space is usually assumed to be one-half the rate of leakage when full of water as during the leakage test.

If w_1 = weight in pounds to fill the clearance space; t minutes = the time required to fill the clearance space; and w_2 = the weight of water in pounds necessary to keep it full for one minute, then the leakage during filling is approximately

$$w' = \frac{w_2}{2} \times t,$$

and the clearance = $(w_1 - w')$ in pounds of water which can be readily reduced to cubic inches.

The clearance for the crank end is found in the same general way as for the head end.

Most engines have small holes at the top of the cylinder at each end (for double-acting engines) which lead into the clearance space. Holes which are covered by the piston on the dead-center would obviously be of no value. All water must of course be drained from each end before filling. Removing the cylinder head for packing the piston is the best method for observing with certainty that there is no water in the head end. The drip pipes in the cylinder can usually be relied on to remove water from the crank end. Determinations should be repeated several times, and the average value is to be used in calculations.

RULES FOR CONDUCTING TESTS OF RECIPROCATING ENGINES. A.S.M.E. CODE of 1912[1]

Determine the object, take the dimensions, note the physical conditions not only of the engine but of all parts of the plant that are concerned in the determinations, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given on pages 258 to 263, and prepare for the test accordingly.

The apparatus and instruments required for a simple performance test of a steam engine, in which the steam consumption is determined by feedwater measurement, are:

(*a*) Tanks and platform scales for weighing water (or water meters calibrated in place).
(*b*) Graduated scales attached to the water glasses of the boilers.
(*c*) Pressure gages, vacuum gages, and thermometers.
(*d*) A steam calorimeter.
(*e*) A barometer.
(*f*) Steam engine indicators.
(*g*) A planimeter.
(*h*) A speed measuring device.
(*i*) A dynamometer for measuring the power developed.

Directions regarding the use and calibration of these appliances are given in the preceding chapters.

The determination of the heat and steam consumption of an engine by feedwater test requires the measurement of the various supplies of water fed to the boiler; that of the water discharged by separators and drips not returned to the boiler, and that of water and steam which escapes by leakage of the boiler and piping; all of these last being deducted from the total feedwater measured.

[1] *Journal of A.S.M.E.*, Nov., 1912.

Where a surface condenser is provided and the steam consumption is determined from the water discharged by the air pump, no such measurement of drips and leakage is required, but assurance must be had that all the steam passing into the cylinders finds its way into the condenser. If the condenser leaks the defects causing it should be remedied, or suitable correction made for the leakage.

To ascertain the consumption of heat, the various feed temperatures are taken and heat calculations made accordingly. If the conditions imposed by the particular method adopted for carrying on the test depart from the usual practice, as for example where a colder supply of feedwater is used than the ordinary supply, a preliminary or subsequent run should be made to ascertain the temperatures which obtain under the usual working conditions, and the heat measurements obtained under the test-conditions appropriately corrected for such departures.

The **steam consumed by steam-driven auxiliaries** which are required for the operation of the engine should be included in the total steam from which the heat consumption is calculated and the quantity of steam thus used should be determined and reported.

Duration. A test for heat or steam consumption, with substantially constant load, should be continued for such time as may be necessary to obtain a number of successive hourly records, during which the results are reasonably uniform. For a test involving the measurement of feedwater for this purpose, **five hours** is sufficient duration. Where a surface condenser is used, and the measurement is that of the water discharged by the air pump, the duration may be somewhat shorter. In this case, successive half-hour records may be compared and the time correspondingly reduced.

When the load varies widely at different times of the day, the duration should be such as to cover the entire period of variation.

The preliminary or subsequent trial for determining the working temperatures on a heat test, where the temperatures obtained under the test conditions depart from the usual temperatures, should be of such duration as may be required to secure working results.

Starting and Stopping. The engine and appurtenances having been set to work and thoroughly heated under the prescribed conditions of test, except in case where the object is to obtain the performance under working conditions, note the water levels in the boilers and feed reservoir, take the time and consider this the starting time. Then begin the measurements and observations and carry them forward until the end of the period determined on. When this time arrives, the water levels and steam pressure should be brought as near as practicable to the same points as at the start. This being done, again note the time and consider it the stopping time of the test. If there are differences in the water levels, proper corrections are to be applied.

Where a surface condenser is used, the collection of water discharged by the air pump begins at the starting time, and the water is thereafter measured or weighed until the end of the test, no observations of the boilers being required.

Care should be taken in cases where the activity of combustion in the boiler furnaces affects the height of water in the gage glasses that the same conditions of fire and drafts are operating at the end as at the beginning. For this reason it is best to start and stop a test without interfering with the regularity of the operation of the feed pump, provided the latter can be regulated to run so as to supply the feedwater at a uniform rate. In some cases where the supply of feedwater is irregular, as, for example, where an injector is used of a larger capacity than is required, the supply of feedwater should be temporarily shut off.

Suitable care should be observed in noting the average height of the water in the glasses, taking sufficient time to satisfactorily judge of the full extent of the fluctuation of the water line, and thereby its mean position.

Records. A set of indicator diagrams should be obtained at intervals of 10 or 20 minutes, and at more frequent intervals if the nature of the test makes it necessary. **Mark on each card the cylinder and the end on which it was taken, also the time of day.** Record on one card of each set the readings of the pressure gages concerned, taken at the same time. These records should subsequently be entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are worked up.

CALCULATION OF RESULTS

(*a*) *Dry Steam.* The quantity of dry steam consumed when there is no superheating is determined by deducting the moisture found by calorimeter test from the total amount of feedwater (the latter being corrected for leakages) or from the amount of air-pump discharge, as the case may be.

When there is superheating the dry steam is found by multiplying the weight of **superheated steam** by the factor

$$1 + \frac{C_p\,(T - t)}{H - q_2}. \qquad (90)$$

in which

C_p = specific heat at constant pressure of superheated steam at observed pressure and temperature.

T = temperature of superheated steam.

t = temperature of saturated steam.

H = total heat of saturated steam at the observed pressure.

q_2 = total heat of feedwater.

(*b*) *Heat Consumption.* The number of heat units consumed by the engine is found by multiplying the weight of feedwater consumed, corrected for leakages, by the total heat of the steam above the working feed temperature, and multiplying the product by a factor of correction expressing the quality of the steam.

If the steam contains moisture, this factor equals

$$x + P\,\frac{q_1 - q_2}{H - q_2}, \qquad (91)$$

in which x is the quality of the steam (one minus the decimal representing the percentage of moisture), P the proportion of moisture, q_1 the total heat of water at the temperature of the steam, q_2 the total heat of the feedwater, and H the total heat of saturated steam at the observed pressure.

If the steam is superheated, the factor is that given above under (*a*) Dry Steam.

If there are a number of sources of feedwater supply, the corresponding heat units should be determined for each supply and the various quantities added together.

The **British standard** of heat consumption is based on a feedwater temperature assumed to be that of the temperature of saturated steam corresponding to the observed back pressure (whether this is above or below the atmosphere), plus the temperature due to heat derived from jacket or reheated drips. It does not include the heat consumed by any auxiliaries, except jackets and reheaters.

(*c*) *Indicated Horse Power.* In a single double-acting cylinder the indicated horse power is found by using the formula

$$\frac{PLAN}{33{,}000}$$

in which P represents the average mean effective pressure in pounds per sq. in. measured from the indicator diagrams, L the length of stroke in ft., A the area of the piston less one-half the area of the piston rod, or the mean area of the rod if it passes through both cylinder heads, in sq. in., and N the number of single strokes of the engine per minute.

Where extreme accuracy is required, the power developed by each side of the piston may be determined and the results added together.

(*d*) *Brake Horse Power.* The brake horse power is found by multiplying the net weight on the brake arm (the gross weight minus the weight when the brake is entirely free) in pounds, the circumference of the circle passing through the bearing point at the end of the brake arm, in ft., and the number of revolutions of the brake shaft per minute; and dividing the product by 33,000.

(*e*) *Electrical Horse Power.* The electrical horse power for a direct-connected generator is found by dividing the output at the bus-bar, expressed in kilowatts, by the decimal 0.746. For alternating-current systems the net output is to be used, being the total output less that consumed for excitation.[1]

(*f*) *Efficiency.* The efficiency is expressed by the thermal efficiency ratio, which is found by dividing the quantity 2545 by the number of heat units consumed per h.p.-hr., either indicated or brake.

(*g*) *Steam accounted for by Indicator Diagrams.* The steam accounted for, expressed in pounds per i.h.p. per hour, may readily be found by using the formula[2]

$$\frac{13{,}750}{\text{m.e.p.}}\left[(C+E)\,W_c-(H+E)\,W_h\right], \quad \ldots \ldots \quad (92)$$

in which

m.e.p. = mean effective pressure, lbs. per sq. in.
C = proportion of stroke completed at cut-off or release.
E = proportion of clearance.
H = proportion of stroke uncompleted at compression.
W_c = weight of 1 cu. ft. steam at cut-off or release pressure.
W_h = weight of 1 cu. ft. steam at compression pressure.

[1] Calculation of electrical output is explained on pages 323 and 324.

[2] Compare with equation (98) page 312.

The points of cut-off release and compression, referred to are indicated in Fig. 299.

In multiple expansion engines the mean effective pressure to be used in the above formula is the combined m.e.p. referred to the cylinder under consideration. In a compound engine the combined m.e.p. for the h.p. (high-pressure) cylinder is the sum of the actual m.e.p. of the h.p. cylinder and that of the l.p. (low-pressure) cylinder multiplied by the cylinder ratio. Likewise the combined m.e.p. for the l.p. cylinder is the sum of the actual m.e.p. of the l.p. cylinder and the m.e.p. of the h.p. cylinder divided by the cylinder ratio.

(h) *Cut-off and Ratio of Expansion.* To find the percentage of cut-off, or what may best be termed the "**commercial cut-off,**" the following rule should be observed:

Through the point of maximum pressure during admission draw a line parallel to the atmospheric line. Through a point on the expansion line, where the cut-off is complete, draw a hyperbolic curve. The intersection of these two lines is the point of commercial cut-off, and the proportion of cut-off is found by dividing the length measured on the diagram up to this point by the total length.

To find the **ratio of expansion** divide the volume corresponding to the piston displacement, including clearance, by the volume of the steam at the commercial cut-off, including clearance.

In a multiple expansion engine the ratio of expansion is found by dividing the volume of the l.p. cylinder, including clearance, by the volume of the h.p. cylinder at the commercial cut-off, including clearance.

TABLE 1. DATA AND RESULTS OF HEAT AND FEEDWATER TESTS OF STEAM ENGINE, SHORT FORM, CODE OF 1912

(1) Test of........................engine located at........................
to determine........................conducted by......................

(2) Type and class of engine and auxiliaries....................................

	1st Cyl.	2d Cyl.	3d Cyl.
(3) Dimensions of main engine:			
(a) Diameter of cylinder....................ins.	......	......	......
(b) Stroke of piston..........................ft.	......	......	......
(c) Diameter of piston rod each end...........ins.	......	......	......
(d) Average clearance....................per cent	......	......	......
(e) Cylinder ratio............................	......	......	......
(f) Horse power constant for 1 lb. m.e.p. and 1 r.p.m.	......	......	......

(4) Dimensions and type of auxiliaries.......................................

(5) Date...

(6) Duration...hrs.

Average Pressures and Temperatures

(7) Pressure in steam pipe near throttle by gage..................lbs. per sq. in.

(7a) Absolute pressure in steam pipe near throttle.................lbs. per sq. in.

(8) Barometric pressure of atmosphere in ins. of mercury......................

(9) Pressure in receivers by gage................................lbs. per sq. in.

(10) Vacuum in condenser in ins. of mercury....................................

(11) Pressure in jackets and reheaters by gage....................lbs. per sq. in.

(11a) Temperature of steam near throttle................................deg. F.

(11b) Temperature of steam in steam chest..........................deg. F.
(12) Temperature of main supply of feedwater..........................deg. F.
(13) Temperature of additional supplies of feedwater....................deg. F.

Total Quantities

(14) Total water fed to boilers from main source of supply....................lbs.
(15) Total water fed from additional supplies..............................lbs.
(16) Total water fed to boilers from all sources..............................lbs.
(17) Moisture in steam or superheating near throttle.............per cent or deg. F.
(18) Factor of correction for quality of steam..
(19) Total dry steam consumed for all purposes..............................lbs.

Hourly Quantities

(20) Water fed from main source of supply......................................lbs.
(21) Water fed from additional supplies...lbs.
(22) Total water fed to boilers per hour..lbs.
(23) Total dry steam consumed per hour.......................................lbs.
(24) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant..lbs.
(25) Net dry steam consumed per hour by engine and auxiliaries...............lbs.
(26) Net dry steam consumed per hour:
(*a*) By engine alone..lbs.
(*b*) By auxiliaries...lbs.

Heat Data

(27) Heat units per lb. of dry steam, based on temperature of Line 12.........B.t.u.
(28) Heat units per lb. of dry steam, based on temperature of Line 13.........B.t.u.
(29) Heat units consumed per hour, main supply of feed.....................B.t.u.
(30) Heat units consumed per hour, additional supplies of feed...............B.t.u.
(31) Total heat units consumed per hour for all purposes....................B.t.u.
(32) Loss of heat per hour due to leakage of plant, drips, etc..................B.t.u.
(33) Net heat units consumed per hour:
(*a*) By engine and auxiliaries..B.t.u.
(*b*) By engine alone..B.t.u.
(*c*) By auxiliaries...B.t.u.

Indicator Diagrams

	1st Cyl.	2d Cyl.	3d Cyl.
(34) Commercial cut-off in per cent of stroke............	......	......	......
(35) Initial pressure in lbs. per sq. in. above atmosphere..	......	......	......
(36) Back pressure at lowest point above or below atmosphere in lbs. per sq. in......................	......	......	......
(37) Mean effective pressure in lbs. per sq. in...........	......	......	......
(38) Steam accounted for by indicator in lbs. per i.h.p. per hour:			
(*a*) Near cut-off..............................	......	......	......
(*b*) Near release..............................	......	......	......

Speed

(39) Revolutions per minute..rev.
(40) Piston speed in ft. per min..ft.

Power

(41) Indicated horse power developed by main engine cylinders:
- 1st cylinder .... i.h.p.
- 2d cylinder .... i.h.p.
- 3d cylinder whole engine .... i.h.p.
- Whole engine .... i.h.p.

(42) Brake horse power .... b.h.p.

Economy Results

(43) Heat units consumed by engine and auxiliaries per hour:
- (*a*) Per indicated horse power .... B.t.u.
- (*b*) Per brake horse power .... B.t.u.

(44) Dry steam consumed per indicated horse power per hour:
- (*a*) By engine and auxiliaries .... lbs.
- (*b*) By main engine alone .... lbs.
- (*c*) By auxiliaries .... lbs.

(45) Dry steam consumed per brake horse power per hour:
- (*a*) By engine and auxiliaries .... lbs.
- (*b*) By main engine alone .... lbs.
- (*c*) By auxiliaries .... lbs.

(46) Percentage of steam used by main-engine cylinders accounted for by indicator diagrams:
- (*a*) Near cut-off .... per cent
- (*b*) Near release .... per cent

(47) Sample Diagrams ....

TABLE 2. DATA AND RESULTS OF STEAM-ENGINE TEST—COMPLETE FORM, CODE OF 1912

(1) Test of .... engine located at ....
to determine .... conducted by ....

(2) Type of engine (simple, compound, or other multiple expansion; condensing or non-condensing) ....

(3) Class of engine (mill, marine, electric, etc.) ....

(4) Rated power of engine ....

(5) Name of builders ....

(6) Number and arrangement of cylinders of engine; how lagged; type of condenser ....

(7) Type of valves ....

(8) Type of boiler ....

(9) Kind and type of auxiliaries (air pump, circulating pump, feed pump; jackets, heaters, etc.) ....

	1st Cyl.	2d Cyl.	3d Cyl.
(10) Dimensions of engine:			
(*a*) Single or double acting	......	......	......
(*b*) Cylinder dimensions:			
Bore, ins	......	......	......
Stroke, ft	......	......	......
Diameter of piston rod, ins	......	......	......
Diameter of tail rod, ins	......	......	......

(*c*) Clearance in per cent of volume displaced by piston per stroke:
Head-end, per cent.................... ....
Crank-end, per cent.................... ....
Average, per cent.................... ....
(*d*) Surface in sq. ft. (average):
Barrel of cylinder, sq. ft.................... ....
Cylinder heads, sq. ft.................... ....
Clearance and ports, sq. ft.................... ....
Ends of piston, sq. ft.................... ....
(*c*) Jacket surfaces or internal surfaces of cylinder heated by jackets, in sq. ft.:
Barrel of cylinder, sq. ft.................... ....
Cylinder heads, sq. ft.................... ....
Clearance and ports, sq. ft.................... ....
Receiver jackets, sq. ft.................... ....
(*g*) Horse power constant for 1 lb. m.e.p. and 1 r.p.m.................... ....

(11) Dimensions of boilers:
(*a*) Number....................
(*b*) Total grate surface....................sq. ft.
(*c*) Total water heating surface....................sq. ft.
(*d*) Total steam heating surface....................sq. ft.

(12) Dimensions of auxiliaries:
(*a*) Air pump....................
(*b*) Circulating pump....................
(*c*) Feed pumps....................
(*d*) Heaters....................

(13) Dimensions of condenser....................
(14) Dimensions of electric or other machinery driven by engine....................
(15) Date....................
(16) Duration....................hrs.

Average Pressures and Temperatures

(17) Steam pressure at boiler by gage....................lbs. per sq. in.
(18) Steam pipe pressure near throttle, by gage....................lbs. per sq. in.
(19) Barometric pressure of atmosphere in lbs. per sq. in....................
(20) Pressure in first receiver by gage....................lbs. per sq. in.
(21) Pressure in second receiver by gage....................lbs. per sq. in.
(22) Vacuum in condenser:
(*a*) In ins. of mercury....................ins.
(*b*) Corresponding absolute pressure....................lbs. per sq. in.
(23) Pressure in steam jacket by gage....................lbs. per sq. in.
(24) Pressure in reheater by gage....................lbs. per sq. in.
(24a) Temperature of steam at boiler....................deg. F.
(24b) Temperature of steam near throttle....................deg. F.
(24c) Temperature of steam in steam chest....................deg. F.
(25) Superheat in steam leaving first receiver....................deg. F.
(26) Superheat in steam leaving second receiver....................deg. F.
(27) Temperature of main supply of feedwater to boilers....................deg. F.
(28) Temperature of additional supplies of feedwater....................deg. F.

(29) Ideal feedwater temperature corresponding to the pressure of the steam in the exhaust pipe, allowance being made for heat derived from jacket or reheater drips (British Standard) .... deg. F.
(30) Temperature of injection or circulating water entering condenser .... deg. F.
(31) Temperature of injection or circulating water leaving condenser .... deg. F.
(32) Temperature of air in engine room .... deg. F.

Total Quantities

(33) Water fed to boilers from main source of supply .... lbs.
(34) Water fed from additional supplies .... lbs.
(35) Total water fed to boilers from all sources .... lbs.
(36) Moisture in steam or superheating near throttle .... per cent or deg. F.
(37) Factor of correction for quality of steam, dry steam being unity ....
(38) Total dry steam consumed for all purposes ....

Hourly Quantities

(39) Water fed from main source of supply .... lbs.
(40) Water fed from additional supplies .... lbs.
(41) Total water fed to boilers per hour .... lbs.
(42) Total dry steam consumed per hour .... lbs.
(43) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant .... lbs.
(44) Net dry steam consumed per hour by engine and auxiliaries .... lbs.
(45) Dry steam consumed per hour:
(*a*) Main engine alone .... lbs.
(*b*) Jackets and reheaters .... lbs.
(*c*) Air pump .... lbs.
(*d*) Circulating pump .... lbs.
(*e*) Feedwater pump .... lbs.
(*f*) Other auxiliaries .... lbs.
(46) Injection or circulating water supplied condenser per hour .... cu. ft.

Heat Data

(47) Heat units per pound of dry steam, based on temperature of Line 27 .... B.t.u.
(48) Heat units per pound of dry steam, based on temperature of Line 28 .... B.t.u.
(49) Heat units consumed per hour, main supply of feed .... B.t.u.
(50) Heat units consumed per hour, additional supplies of feed .... B.t.u.
(51) Total heat units consumed per hour for all purposes .... B.t.u.
(52) Loss of heat per hour due to leakage of plant, drips, etc .... B.t.u.
(53) Net heat units consumed per hour:
(*a*) By engine and auxiliaries .... B.t.u.
(*b*) By engine alone .... B.t.u.
(*c*) By auxiliaries .... B.t.u.
(54) Heat units consumed per hour by the engine alone, reckoned from temperature given in Line 29 (British Standard) .... B.t.u.

Indicator Diagrams

	1st Cyl.	2d Cyl.	3d Cyl.
(55) Commercial cut-off in per cent of stroke	......	......	......
(56) Initial pressure in lbs. per sq. in. above atmosphere ..	......	......	......
(57) Back-pressure at mid-stroke above or below atmosphere in lbs. per sq. in	......	......	......

(58) Mean effective pressure in lbs. per sq. in. . . . .
(59) Equivalent mean effective pressure in lbs. per sq. in.:
(*a*) Referred to first cylinder . . . .
(*b*) Referred to second cylinder . . . .
(*c*) Referred to third cylinder . . . .
(60) Pressures and percentages used in computing the steam accounted for by the indicator diagrams, measured to points on the expansion and compression curves:
Pressure above zero in lbs. per sq. in.:
(*a*) Near cut-off . . . .
(*b*) Near release . . . .
(*c*) Near beginning of compression . . . .
Percentage of stroke at points where pressures are measured:
(*d*) Near cut-off . . . .
(*e*) Near release . . . .
(*f*) Near beginning of compression . . . .
(61) Aggregate m.e.p. in lbs. per sq. in. referred to each cylinder given in heading . . . .
(62) Mean back pressure above zero, lbs. per sq. in . . . .
(63) Steam accounted for in lbs. per indicated horse power per hour:
(*a*) Near cut-off . . . .
(*b*) Near release . . . .
(64) Ratio of expansion . . . .
(65) Mean effective pressure of ideal diagram[1] . . . . lbs. per sq. in.
(66) Diagram factor[1] . . . .

Speed

(67) Revolutions per minute . . . .
(68) Piston speed per minute . . . . ft.
(69) Variation of speed between no load and full load . . . . r.p.m.
(70) Fluctuation of speed on suddenly changing from full load to no load, measured by the increase in the revolutions due to the change . . . . r.p.m.

Power

(71) Indicated horse power developed by main engine:
1st cylinder . . . . i.h.p.
2d cylinder . . . . i.h.p.
3d cylinder . . . . i.h.p.
Whole engine . . . . i.h.p.
(72) Brake horse power . . . . b.h.p.
(73) Friction i.h.p. by diagrams, no load on engine, computed for average speed . . i.h.p.
(74) Difference between Lines 71 and 72 . . . . h.p.
(75) Percentage of i.h.p. of main engine lost in friction . . . . per cent
(76) Power developed by auxiliaries[2] . . . . i.h.p.

[1] See *Journal of A.S.M.E.*, Nov., 1912, page 1861.
[2] These are not included in the power developed by the main engine.

Economy Results

(77) Heat units consumed per indicated horse power per hour:[1]
- (a) By engine and auxiliaries.......B.t.u.
- (b) By engine alone.......B.t.u.

(78) Heat units consumed per brake horse power per hour:
- (a) By engine and auxiliaries.......B.t.u.
- (b) By engine alone.......B.t.u.

(79) Heat units consumed by engine per hour, corresponding to ideal temperature of feedwater given in Line 29, per indicated horse power (British Standard).......B.t.u.

(80) Dry steam consumed per i.h.p. per hour:
- (a) By engine and auxiliaries.......lbs.
- (b) By main engine alone.......lbs.
- (c) By auxiliaries.......lbs.

(81) Dry steam consumed per brake h.p. per hour:
- (a) By engine and auxiliaries.......lbs.
- (b) By main engine alone.......lbs.
- (c) By auxiliaries.......lbs.

(82) Percentage of steam used by main engine cylinders accounted for by indicator diagrams:

	1st Cyl.	2d Cyl.	3d Cyl.
(a) Near cut-off.......	......	......	......
(b) Near release.......	......	......	......

Efficiency Results

(83) Thermal efficiency ratio for engine and auxiliaries:
- (a) Per indicated horse power.......per cent
- (b) Per brake horse power.......per cent

(84) Thermal efficiency ratio for engine alone:
- (a) Per indicated horse power.......per cent
- (b) Per brake horse power.......per cent

(85) Ratio of economy of engine to that of an ideal engine working with the Rankine cycle (see page 308).......per cent

Work Done per Heat Unit

(86) Ft.-lbs. of net work per B.t.u. consumed by engine and auxiliaries (1,980,000 ÷ Line 78a).......ft.-lbs.

Note. Both the Short Form and Complete Form here given refer to a steam engine used for general service.

For an engine driving an electric generator the form should be enlarged to include the electrical data, embracing the average voltage, number of amperes in each phase, number of watts, number of watt hours, average power factor, etc.; and the economy results based on the electrical output embracing the heat units and steam consumed per electric h.p. per hour and per kw.-hr., together with the efficiency of the generator. See table for Steam Turbine Code, pages 325 to 328.

Likewise, in a marine engine having a shaft dynamometer, the form should include the data obtained from this instrument, in which case the brake h.p. becomes the shaft h.p.

[1] The h.p. on which the economy and efficiency results are based are those of the main engine given in Line 71.

Surface condensers are usually perferred for accurate engine tests because the steam used by the engine can be determined directly by weighing or measuring the condensed steam. A surface condenser is essentially a vessel of considerable size in which there are a great many brass tubes. It is usually designed so that exhaust steam passes on its way through the condenser into contact with the outside surface of these tubes while cold water for condensing the steam circulates inside the tubes. Steam condensed in this way accumulates in the bottom of the condenser and is removed by an air pump used for producing a vacuum, or by gravity if the pressure in the condenser is atmospheric, as it will be if the engine is operating "non-condensing," that is, without a vacuum. It is very essential, however, that **surface condensers be tested for leakage,** preferably before and after every important test is made; and the leakage should be determined with the same vacuum in the condenser as there is when it is used in the engine test. Probably the best method to determine the amount of condenser leakage is to pass cooling water through the tubes at the normal rate of flow, maintaining at the same time with the air pump the required vacuum. Then the amount of water removed by the air pump is the leakage of circulating water through the tubes into the steam space. Under normal operating conditions there would be **no leakage of steam into the circulating water,** because the water will be at a higher pressure than the steam.

If only a test is to be made to determine whether or not there is any leakage in the condenser, the test most generally applied is to close all connections to the condenser and observe whether a vacuum once established can be maintained a reasonably long time. A more rapid test applicable, however, only where salt water is used for cooling, is to test the condensed steam several times during a test by adding a little silver nitrate to a small amount of the condensed steam. If there is no appreciable precipitate[1] it may be assumed that the condenser does not leak.

Some of the important considerations to be observed in an accurate engine test will now be given as stated in the rules adopted by the American Society of Mechanical Engineers.

Discussion of Thermal Efficiency. The ratio of the heat converted into work to the heat supplied to the engine is called the thermal efficiency. The first of these quantities is calculated from the indicated horse power and the latter from the weight and total heat per pound of the steam supplied. The only uncertainty arises as to the proper **base** from which to calculate. The total heat of steam given in steam tables is calculated from 32 degrees Fahrenheit, but it is obvious if this **base** is adopted the

[1] The white precipitate formed with the salt in sea water is of course silver chloride, thus, $AgNO_3 + NaCl = AgCl + NaNO_3$.

engine may be charged with more than its share of heat. If, for example, the exhaust steam from the engine passes through a feedwater heater and the engine returns the condensed steam to the boiler as feedwater at say 150 degrees Fahrenheit, then there will be 150 − 32, or 118 B.t.u. in every pound of steam passing continually from the engine to the boiler and from the boiler back to the engine without doing any work. More accurately, then, the thermal efficiency of an engine (E_t) should be stated as

$$E_t = \frac{Q_u + Q_f}{Q_n}, \qquad (93)$$

where Q_u and Q_f are the heat equivalents respectively of the useful work and of the engine friction, while Q_n should be defined as the **net heat supplied** to the engine. From this discussion it follows that the more efficient the feedwater heater the higher the efficiency of the engine will be, and if an efficient heater is not used there is no reason for charging a loss to the engine. Obviously the limit to the amount of heat returnable to the boiler by means of a heater is the amount of heat in the exhaust steam, and this limit can be nearly approached under actual practical conditions. It is, therefore, very reasonable that the " datum " for the calculation of the heat supplied to the engine should be the heat of the liquid corresponding to the temperature of the steam in the exhaust pipe from the engine. All this applies, of course, equally well to engines whether operating condensing or non-condensing. In other words, the net heat supplied to the engine is the total heat of the steam entering the engine, less the heat of the liquid at the temperature of the engine exhaust.

The temperature of the exhaust must be taken by a thermometer in a suitable thermometer cup placed in the **exhaust pipe** close to the engine; and, similarly, the temperature and quality of the steam supplied to the engine must be determined by thermometers and a steam calorimeter placed close to, but on the boiler side of, the engine throttle valve.

Fig. 303 shows diagrammatically the heat distribution and various losses in a steam plant, and gives approximate percentage values of the losses.

Engine Performance Expressed in British Thermal Units. A method which is rapidly gaining in favor among practical engineers is to express the performance of a steam engine or turbine by the number B.t.u. supplied per indicated horse power per minute, the heat units supplied per minute being determined, as explained in the preceding paragraphs; that is, the total heat in the steam entering at the throttle less the heat of the liquid at the temperature of the exhaust.

Hirn's Analysis. For certain scientific investigations it is useful to make a heat analysis of the indicator diagram, to show the interchange of heat from steam to cylinder walls, etc., which is going on within the

FIG. 303. — Heat Distribution and Losses in a Steam Boiler and Engine Plant (Parsons).

cylinder. This is unnecessary for commercial tests. Years ago deductions from such an analysis were considered to be of considerable importance to designers; but, lately, such data are considered of very doubtful importance.

Ratio of Economy of an Engine to that of an Ideal Engine. The cycle of the ideal engine recommended for obtaining this ratio is that which was adopted by the Committee appointed by the *Civil Engineers* of London, to consider and report a standard thermal efficiency for steam engines. This engine is one which follows the **Rankine cycle** where steam at a constant pressure is admitted into a cylinder having no clearance, and after the point of cut-off is expanded adiabatically to the back-pressure. In obtaining the economy of this engine the **feed-water is assumed to be returned to the boiler at the exhaust temperature (Fig. 304).**

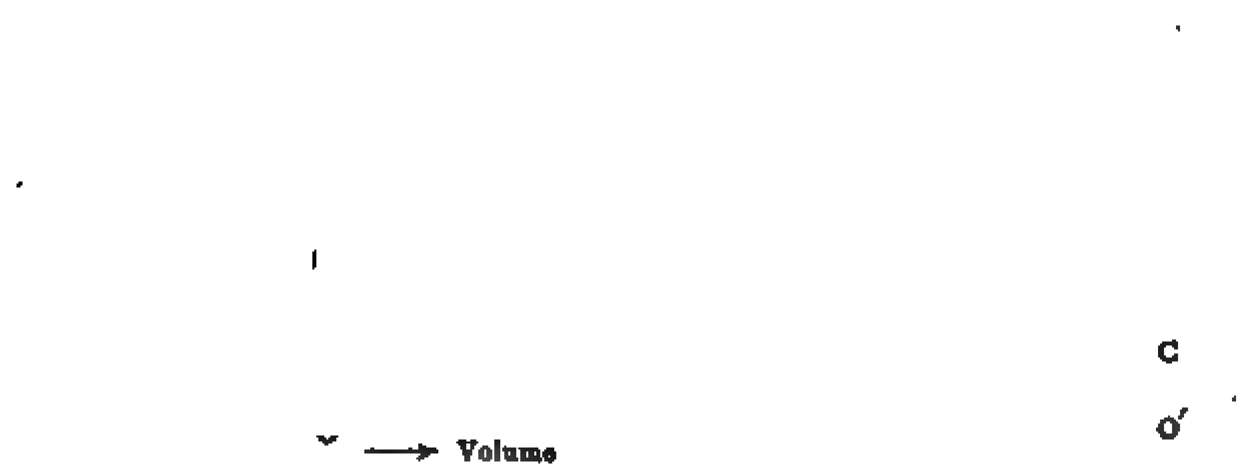


FIG. 304. — Indicator Diagram for the Ideal Rankine Cycle.

The ratio of the economy of an engine to that of the ideal engine is obtained by dividing the heat consumption per indicated horse power per minute for the ideal engine (called the " theoretical water-rate ") by that of the actual engine.

Engine Performance Compared with the Rankine Cycle. In order to know how much the efficiency of an engine can be improved it is most desirable to compare the actual thermal efficiency, with the highest possible efficiency. For steam engines the standard cycle for comparison is now generally taken as the Rankine cycle,[1] in which the operation of the engine is assumed to be perfect; that is, without clearance in the cylinder, initial condensation, leakage or radiation. The indicator diagram for the Rankine cycle is represented by **Fig. 304.** The steam is assumed to be supplied to the engine cylinder at constant pressure until the point of cut-off, after which it is expanded **adiabatically** down to the

[1] Years ago it was not unusual to make this comparison with the efficiency of the Carnot cycle as a basis. This efficiency of a heat engine, it will be remembered, is expressed by the ratio of $T_1 - T_2$ to T_1 where T_1 is the absolute initial temperature and T_2 is the absolute final temperature.

back pressure at which the engine is operated on the return stroke when the exhaust steam is swept out of the cylinder and returned as feed-water to the boiler at the temperature of the exhaust. The same Rankine cycle represented in **Fig. 304** when shown by a so-called **entropy-temperature diagram** can be made simpler both for analysis and calculations. This other kind of diagram, the details of which are somewhat more difficult to understand, is universally used by steam turbine engineers and has for the problem in hand particular advantages. In this diagram, any surface represents accurately to given scales a quantity of heat. Absolute temperatures (T) are the ordinates, and entropies[1] (ϕ) are the abscissas.

Specific Heat of Superheated Steam. In modern practice superheated steam often enters our calculations. The specific heat of steam

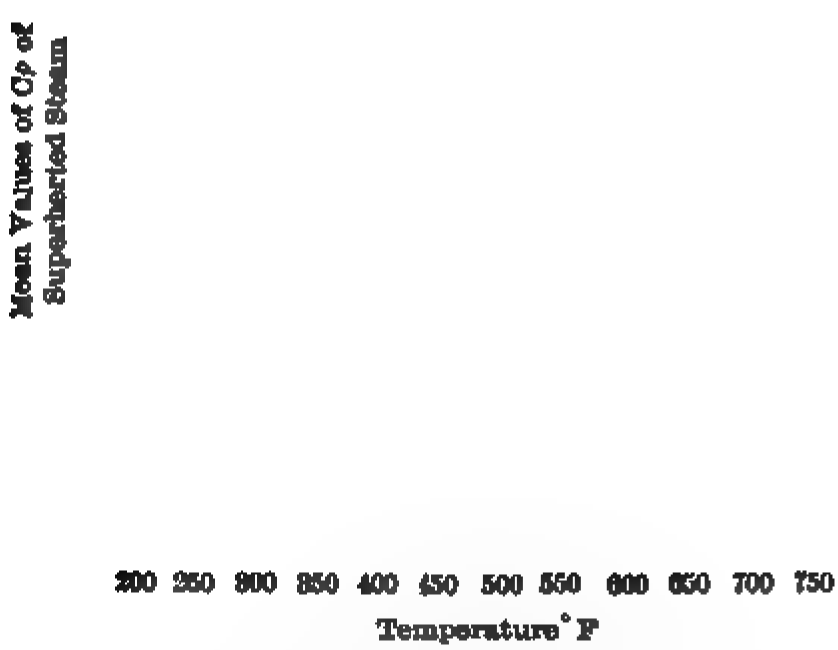


FIG. 305. — Mean Values of Specific Heat (C_p) of Superheated Steam Integrated from Knoblauch and Jacob's Data.

varies with the temperature and pressure as shown in **Figs. 305 and 306**, giving values of the **mean** and the **true** specific heat at constant pressure (C_p), as determined by Knoblauch and Jakob.[2]

[1] Entropy, which Perry calls the "ghostly quantity," has no real physical significance, so that complete definition is not possible. If dQ is a small amount of heat added to a body and T is the absolute temperature at which the heat is added, then the change in entropy of that body is dQ/T, or $d\phi = dQ/T$. Entropy of saturated steam above the entropy of water at the freezing point is easily calculated. For saturated steam at any pressure, then $\phi = xr/T + \theta$, where x is the quality of the steam, r is the heat of vaporization, T is the absolute temperature, and θ is the entropy of the liquid (water).

[2] *Zeit. Verein deutscher Ingenieure*, Jan. 5, 1907. Values of mean specific heat are taken from *Mechanical Engineer*, July, 1907, and Professor A. M. Greene's paper in *Proc. American Society of Mechanical Engineers*, May, 1907.

True specific heat represents the ratio of the amount of heat to be added to a given weight of steam at some particular condition of temperature and pressure to raise the temperature one degree to that required to raise the temperature of water at maximum density one degree. The **mean specific heat** is almost invariably used in steam engine and turbine calculations.

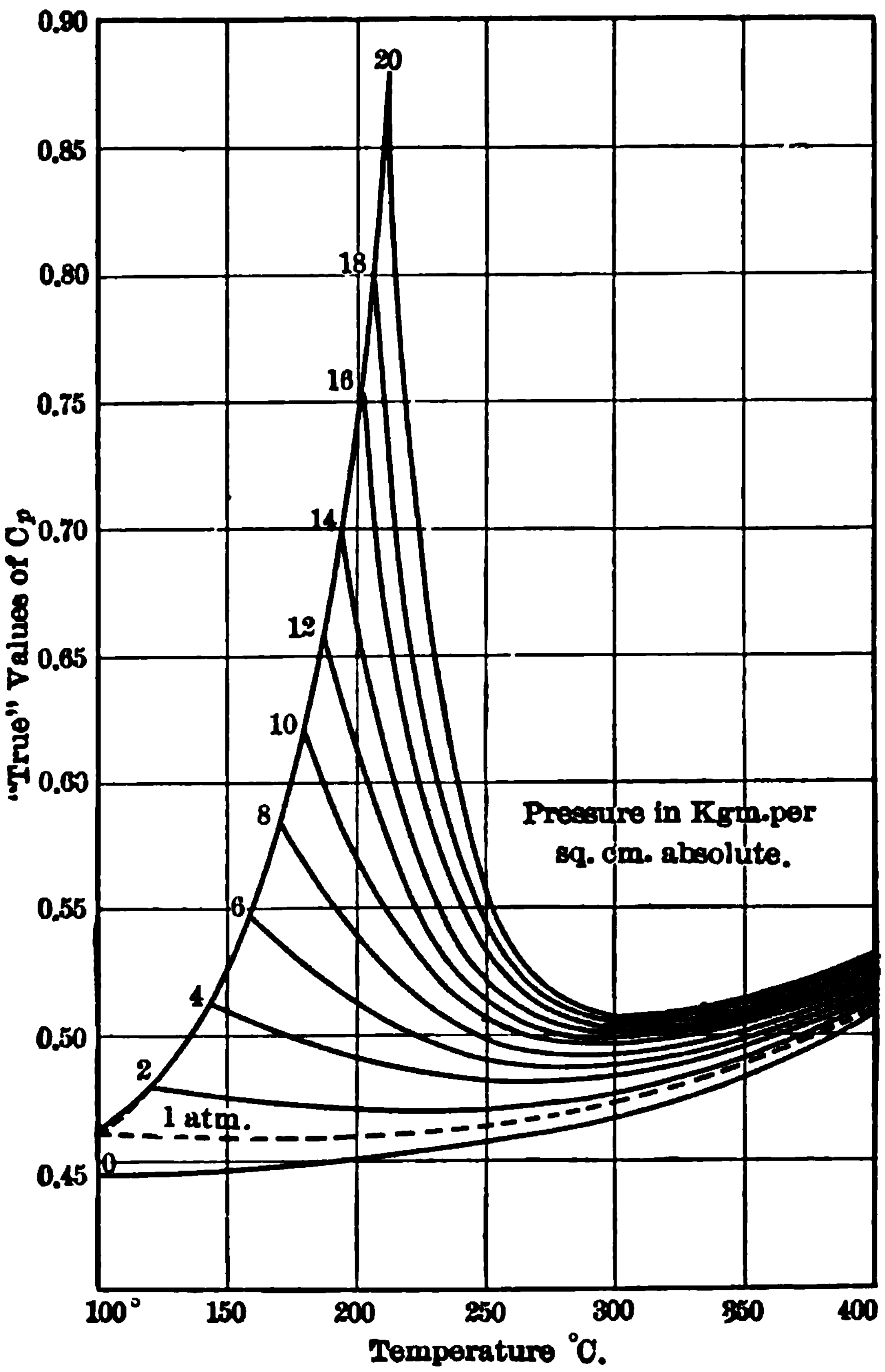


FIG. 306. — Values of the "True" Specific Heat of Superheated Steam.

Approximate Steam Consumption Calculated from an Indicator Diagram. It is often very convenient to be able to calculate the approximate steam consumption of a steam engine from the data obtainable from an indicator card, the size of the piston, the stroke, and the speed. Using a double-acting engine, the following symbols[1] may be used:

[1] Compare with *Power*, September, 1893.

p = mean effective pressure, pounds per square inch from indicator diagram.

l = length of the stroke of the engine in feet.

a = area of the piston in square inches.

b = percentage of clearance to the length of the stroke.

c = percentage of stroke at any point in the expansion line.[1]

n = number of revolutions per minute; and 120 n = number of strokes per hour.

w = weight of a cubic foot of steam having a pressure as shown by the indicator diagram corresponding to that at the point in the expansion line selected for c, pounds.

w′ = weight of a cubic foot of steam corresponding to the pressure at the end of compression, pounds.

Then the number of cubic feet per stroke $= \frac{la\,(b + c)}{144\,(100)}$ in the clearance and piston displacement volumes (at c).

$$\text{Weight of steam per stroke, pounds} = \frac{law\,(b + c)}{144\,(100)}. \quad . \quad . \quad . \quad . \quad (94)$$

$$\text{Volume of the clearance, cubic feet} = \frac{la\,(b)}{144\,(100)}.$$

Weight of steam in clearance, pounds remaining in the cylinder

$$= \frac{law'\,(b)}{144(100)}.$$

Approximate net weight of steam used per stroke

$$= \frac{law\,(b + c)}{144\,(100)} - \frac{law'\,(b)}{144\,(100)} = \frac{la}{14{,}400}\,[(b + c)\,w - bw']. \quad . \quad (95)$$

Approximate weight of steam from diagram per hour

$$= \frac{120\,nla}{14{,}400}\,[(b + c)w - bw']. \quad . \quad . \quad . \quad . \quad . \quad (96)$$

Indicated horse power for a double-acting engine

$$= \frac{2\,plan}{33{,}000}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (97)$$

[1] In other words this is the percentage of the entire stroke which has been swept through by the piston corresponding to the point in the expansion curve selected for measurements. It is preferable, however, to take this point not very far from the point of cut-off, since the assumption must be made that the product of pressure times volume in the expansion curve is constant, which is, of course, not accurate, and the error becomes greater as the expansion increases.

Steam consumption per indicated horse power[1] is (96) divided by (97) or

$$= \frac{137.5}{p}\left[(b + c)\,w - bw'\right]. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (98)$$

Compare this with formula (92) page 297.

The difference between the theoretical steam consumption calculated by the formula and the actual consumption as determined by tests represents steam "not accounted for by the indicator," due to cylinder condensation, leakage through ports, radiation, etc. If the steam supplied to the engine is very wet, corrections for this moisture should be made in the value of **w**.

Willans Law. One of the most serviceable **checks** that can be applied to engine tests is plotting the **Willans line** of total steam consumption per hour. Curve sheets illustrating this as plotted from actual tests by Barraclough and Marks[2] are shown in **Fig. 307**. It will be observed that the points representing the weight of steam used per hour when plotted for the horse power corresponding are on a **straight line.** In other words Willans law is usually stated thus: "With a fixed cut-off and a **throttling governor** the total steam used by the engine per hour at different loads can be represented by a straight line upon a mean effective pressure base or upon a horse-power base." It will be shown also in the following paragraphs how, theoretically, this relation holds for an engine operating at a fixed cut-off and with a throttling governor and that the total steam used per hour is proportional to the mean effective pressure and also to the horse power developed.

0 10 20 30
Indicated Horse Power

FIG. 307. — "Willans" Lines for an Engine with a Throttling Governor.

[1] A method of determining steam consumption by means of logarithmic curves of indicator diagrams by J. P. Clayton is described in Bulletin No. 65 of Engineering Experiment Station of University of Illinois and in abridged form in *Journal of A.S.M.E.*, April, 1912.

[2] *Proceedings Institution of Civil Engineers*, vol. 120, page 323.

If we assume that the expansion curve is hyperbolic, which is usually near the truth, then the mean "forward" pressure given by an indicator diagram is[1]

$$p_1\left(\frac{1 + \log_e r}{r}\right), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (99)$$

where p_1 is the initial pressure of the steam and r is the ratio of expansion. With a **throttling** governor r is of course constant. The terms in the parenthesis can then be represented by a constant c and the mean forward pressure p_f then can be written as p_1c. Volume of steam used per hour for a **double-acting** engine can be expressed in cubic feet as 120 (nV), where n is the number of revolutions per minute and V is the volume of steam admitted to the cylinder per stroke. Now if we use the symbol v_1 for the specific volume, that is, the volume in cubic feet of a pound of steam at the pressure p_1, and assuming for a small range of pressures that p_1v_1 = a constant k, then we can write, if W is the weight of steam used per hour in pounds,

$$W = \frac{120\,(nV)}{v_1} = \frac{120\,(nVp_1)}{k} \quad . \quad . \quad . \quad . \quad . \quad . \quad (100)$$

Now with a throttling governor and constant cut-off all these quantities are constant except p_1, and writing a constant z for the term $\frac{120\,(nV)}{k}$ we have $W = zp_1$, but it was shown above that the mean **forward** pressure $p_f = cp_1$, so that $W = z\frac{p_f}{c}$.

So that the curve representing this equation is a **straight line** and passes through the origin of coördinates. If, however, we use the mean **effective** pressure instead of the mean forward pressure, then

$$\text{m.e.p.} = p_f - p_b,$$

where p_b is the mean "back" or exhaust pressure. In these last terms then

$$W = \frac{z}{c}(\text{m.e.p.} + p_b). \quad . \quad . \quad . \quad . \quad . \quad . \quad (101)$$

This last equation may be stated as W = a constant $\times$ m.e.p. + another constant which, when plotted to a scale of mean effective pressure for abscissas and weight of steam used per hour for ordinates, is also a **straight line**, intersecting the axis of ordinates above the origin at a distance corresponding to the steam consumption per hour at no load.

Since the **indicated horse power** (i.h.p.) is proportional to mean effective pressure, a straight line will result when the steam consumption and indicated horse power are plotted; and the same holds true also

[1] Compare with Perry's "Steam Engine," page 286.

when steam consumption is plotted with **brake horse power (b.h.p.)** instead of indicated horse power.

Curves Showing Results of Tests Graphically. One of the best checks of an engine test is to plot the principal observations graphically as the test proceeds. This is particularly important as regards the **total weight of steam used** per hour. For a series of tests each made with a different load the points plotted with horse power (either indicated or brake) as abscissas and total steam per hour as ordinates should lie along a **straight line,** known as the **Willans line.** This statement applies accurately only for engines operating with a throttling governor at, of course, constant speed, but is generally applicable to steam turbine tests, irrespective of the type of governor.

CHAPTER XIII

TESTING STEAM TURBINES AND TURBO-GENERATORS

Testing Steam Turbines.[1] In every power plant the means should be available for making tests of the steam equipment to determine the steam consumption. Usually tests are made to determine how nearly the performance of a turbine approaches the conditions for which it was designed. The results obtained from tests of a turbine are to show usually the steam consumption required to develop a unit of power in a unit of time, as, for example, a horse power or a kilowatt-hour.

In such tests a number of observations must be made regarding the condition of the steam in the passage through the turbine and of the performance of the turbine as a machine. To get a good idea of what these observations mean, it may be profitable to follow the steam as it passes through the turbine. The steam comes from the boilers through the main steam pipe and the valves of the turbine to the nozzles or stationary blades as the case may be. It then passes through the blades and finally escapes through the exhaust pipe to the condenser. It is preferable to have a surface condenser for tests, so that the exhaust steam can be weighed. The weighing is preferably done in large tanks mounted on platform scales.

Methods for Testing. The important observations to be made in steam turbine tests are:

1. Pressure of the steam supplied to the turbine.

2. Speed of rotation of the turbine shaft, usually taken in revolutions per minute.

3. Measurement of power with a Prony or a water brake, if the power at the turbine shaft is desired; or with electrical instruments (ammeters, voltmeters, and wattmeters), if the power is measured by the output of an electric generator.

4. Weight, or measurement by volume, of the condensed steam discharged from the condenser. Unless a surface condenser is used it is very

[1] Tests of the turbines alone in a modern station may be only a rough indication of the over-all economy of the plant. Recently steam turbines were installed in a large power plant where they replaced steam engines of an excellent make. Tests of the turbines and of the engines made without considering the losses in the rest of the plant showed very little gain in efficiency by this change, although it was found that the fuel consumption was reduced twenty per cent.

Parts of this chapter and the chapter following are taken from the author's work on "*The Steam Turbine*" (John Wiley & Sons, New York).

difficult to obtain the amount of steam used by the turbine. All leakages from pipes, pumps, and valves, which are a part of the steam which has gone through the turbine, must be added to the weight of the condensed steam. The accuracy of a test often depends a great deal on how accurately leaks have been provided against, or measured when they occur.

5. Temperature of the steam as it enters the turbine.[1]

6. Vacuum or back-pressure in the exhaust pipe of the turbine.

All gages, electrical instruments, and thermometers should be carefully calibrated before and after each test, so that observations can be corrected for any errors. The zero readings of Prony and water brakes for measuring power should be carefully observed and corrected to eliminate the friction of the apparatus with no load. Unless all these precautions are taken the difficulties in getting reliable tests of turbines are greatly increased. In all cases tests should be continued for several hours with absolutely constant conditions if the tests are to be of value.

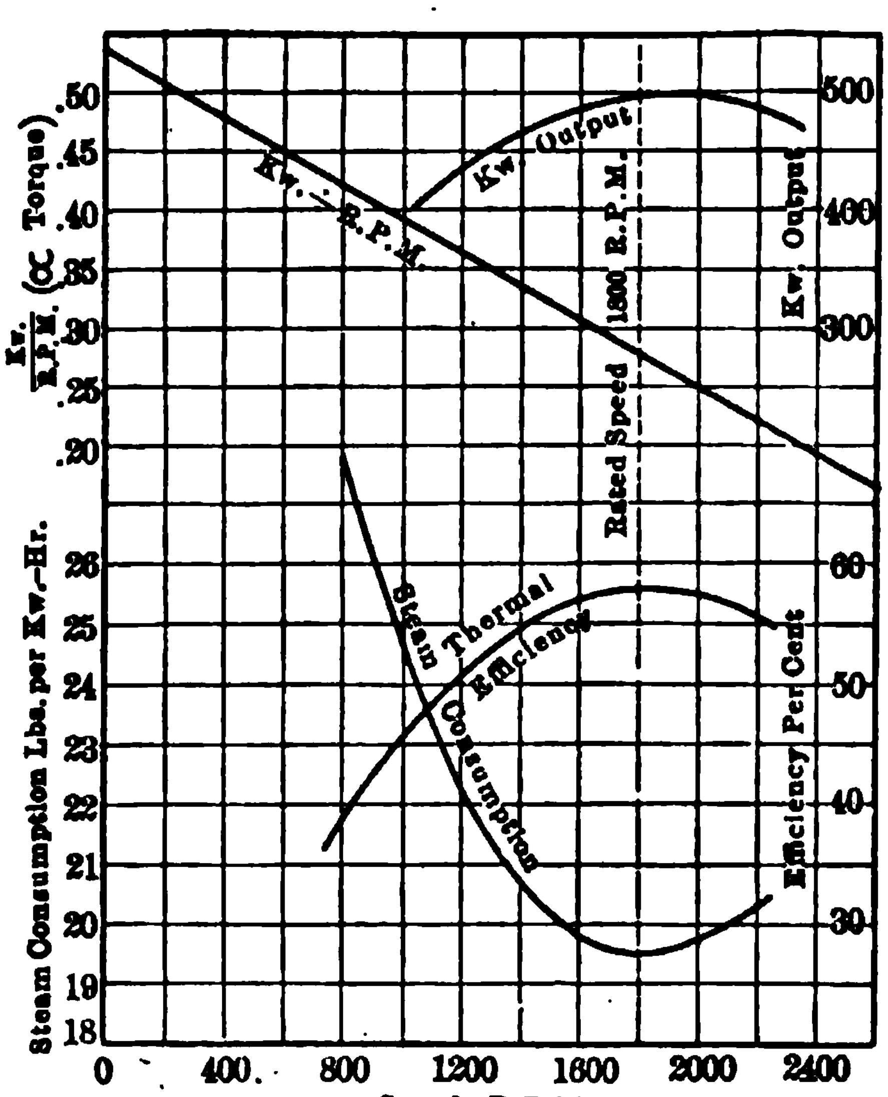


Fig. 308. — Results of Tests of a Turbine at Various Speeds.

The most valuable test of a steam turbine or of a reciprocating steam engine is made when varying only the load; that is, with pressures, superheat, and speed constant. When the steam consumption is then plotted against fractions of full load, a **water-rate curve** is obtained. For such a curve a series of tests are needed, each for some fraction of full load; and in each separate test the power as well as all the other conditions must be held constant.

Another important test of the performance of a steam turbine is made by varying both the speed and the power and keeping the other conditions constant. The observations of speed and power from such a test give a power parabola as illustrated in **Fig. 308**. This curve shows at what speed the turbine gives the greatest output.

[1] The most satisfactory tests of turbines are made with steam slightly superheated rather than wet. When steam is very wet (more than about 4 per cent moisture for ordinary pressures) the determination of the quality is difficult. There is also a danger that steam showing only a few degrees of superheat by the reading of the thermometer is actually wet. The high temperature is due in such cases to heating from eddies around the thermometer case or in steam pockets near it.

Tests may also be made with varying initial steam pressure, but keeping other conditions including exhaust pressure and load constant.

Calculations of the steam consumption and efficiency of turbines made by allowing for the different losses as calculated separately and then added together as is often done to determine the losses in electrical apparatus are of very little value except when made by experienced designers.[1]

Commercial Testing. The methods used by the New York Edison Company in commercial tests of steam turbine-generator units may well be explained briefly.

During a test the load on the turbine unit is maintained as constant as possible by "remote control" of the turbine governor by the switchboard operator. The maximum variation in load is to be held within 4 per cent above and below the mean. For some time previous to the test the turbine is run a little below the load required for the test, but at least ten minutes before the starting signal is given the test load must be on the machine.

Three-phase electrical load is measured by the two-wattmeter method,[2] using Weston indicating wattmeters of the standard laboratory type. These instruments are calibrated at a well-known testing laboratory immediately before and after the test. Power factor is maintained substantially at unity and all electrical readings are taken at one-minute intervals.

When the turbine is supplied with a surface condenser, the steam consumption, or water rate, is determined by weighing in a large tank supported on platform scales the condensed steam delivered from the condenser hot well. Above the tank on the scales a reservoir is provided which is large enough to hold the condensation accumulating between the weighings, which were made at intervals of five minutes. By using a loop connection for the gland water supply (of Westinghouse turbines) or the water from the step bearing (of Curtis turbines, using water for this bearing) the necessity for correcting the weighings for these amounts is avoided.

Because the circulating water at the stations of this company is usually quite salty, any condenser leakage is detected by testing the condensed steam by the silver-nitrate test with a suitable color indicator. This color method is said to be a decided advantage over the usual method of weighing the leakage accumulating during a definite period when the

[1] This method of calculation of steam consumption is explained in detail in "The Steam Turbine," by the author, pages 86–93. Steam consumption of a turbine can be predicted by calculations much more accurately than for a steam engine.

[2] See Kent's "Mechanical Engineer's Pocket-Book," 8th Ed., page 1396, or Foster's "Electrical Engineer's Pocket-Book," 4th Ed., pages 51 and 325.

condenser is idle and tested only with full vacuum. By taking samples of circulating water and condensed steam at the same time, it is possible to detect any change in the rate of condenser leakage.

The water level in the hot well is maintained at practically a constant point by means of a float valve in the well, automatically controlling the speed and, therefore, the amount of the delivery of the hot-well pump. This device avoids the necessity for the difficult correction to be made in a test when the levels in the hot well are not the same at the beginning and end of a test. Temperatures and pressures of the admission steam are determined by mercury thermometers and pressure gages located near the main throttle valve of the turbine; the amount of superheat is determined by subtracting from the actual steam temperature after making thermometer connections the temperature of saturated steam corresponding to the pressure at the point where the temperature is measured. All gages and thermometers are calibrated before and after the test.

Vacuum is measured directly at the turbine exhaust by means of a mercury column with a barometer alongside for reducing the vacuum to standard barometer conditions (30 inches). By this latter arrangement the necessity for temperature corrections between the two mercury columns not at the same place is avoided.

It has sometimes happened that split condenser tubes have caused a leakage of steam which was extremely difficult to measure. Cases are reported where the split opened up only when the condenser was heated with a large volume of steam. On this account it is preferable not to use a leaky condenser for accurate tests; in other words, the condenser should be thoroughly repaired before tests are made. The effect of split tubes causing an irregular amount of leakage is usually shown in tests by inconsistent results in the weight of condensed steam. In that case the leakage will be greatest with largest flow of steam through the condenser and it will be observed, for an engine operating with a throttling governor or with a steam turbine, that when the "Willans line" page 274, is drawn to check the tests that it will be curved instead of straight. It should be noted, however, that a curved "Willans line" does not necessarily indicate this phenomenon in condenser leakage, as the irregularity may be due to faulty design of the engine[1] or turbine. Tests of condensers for leakage should be run long enough so that the quantity of water coming through can be determined with accuracy.[2] Usually a leakage test run for less than an hour or two is of no use at all.

[1] The "Willans" line for a reciprocating engine operating with an automatic cut-off governor is usually a curve slightly concave upward.

[2] To determine accurately the weight of condensed steam the air-pump, piping, and tanks must be free from leaks, and the condenser and pump should be so arranged

When the method of determining the weight of condensed steam by weighing the boiler feed-water is used the chances of error are very great and every possible precaution to insure accuracy must be observed. In the first place valves no matter how good should not be relied on to prevent the passage of steam through them. For this reason careful engineers insist on disconnecting from the line of steam piping between the boilers and the engine or turbine tested all other piping connected to it, and then blanking off with flanges all the sections disconnected. If flanged pipe fittings have been used in the pipe lines, blanking off sections in the various pipes is very easily accomplished by disconnecting the flanges and inserting a thin iron or copper plate with holes around the edge to fit the bolt-holes in the flanges. The plate is then easily bolted in place. Another important precaution to observe is that the outlets of all drain or drip pipes and of all blow-off valves must be visible. It is equally important that all the piping between the boiler feed-pumps and the boilers is exposed with all branches blanked off or plugged. Boiler leakage should be determined before and after each test with preferably the pipe supplying the turbine blanked off at the throttle valve, although if the throttle valve is reasonably tight the precaution of blanking off this valve is not considered so important as the others mentioned. In tests for boiler leakage the required steam pressure must be maintained on both the piping and the boilers. Measuring feed-water with water meters should not be thought of unless only approximate results are expected, and in that case such an arrangement is only allowable if the temperature of the boiler feed is not over 80 to 90 degrees Fahrenheit for most meters, and a by-pass at the meter is provided, so that at frequent intervals the meter can be calibrated by actual weighing of the flow through it, with the rate of flow and temperature of the water the same as in the tests. Boiler leakage is often as much as from ten to fifteen per cent of the weight of feed-water, and in some reliable tests a still greater leakage has been observed.

Steam Consumption Determined by a "Heat Balance" Method. There is still another method sometimes used for determining the steam consumption of engines and turbines operated with jet condensers or condensers of a similar type where the cooling water and condensed steam are mixed and discharged together from the condenser. This method is based on the measurement of the amount of heat absorbed by the cooling water from the condensed steam. The weight w_c and temperature of the cooling water leaving the condenser t_2, the quality of the exhaust steam x and the temperature of the mixture of condensed steam and cooling water t'' are determined as accurately as possible and from

with respect to each other that the condensed steam will flow in a continuous stream to the pump and into the tanks.

these data the weight of condensed steam w_s is of course readily calculated by a simple algebraic equation as follows:

$$w_c (t_2 - 32) + w_s (q + xr) = (w_c + w_s)(t'' - 32), \quad . \quad . \quad (102)$$

where q and r are respectively the heat of the liquid and the heat of vaporization corresponding to the temperature of the exhaust steam. In this equation " heat contents " are measured for each term from 32 degrees F. The method is, however, unreliable, and at best can be depended on for only very approximate results. The reason for this inaccuracy is the difficulty of measuring, especially in large plants, the quantity of cooling water and the true average temperature of a large volume of water flowing in a pipe or channel. It is found usually that the temperature of the water discharged from the condenser will vary from one side of the pipe to the other, and small errors in the determination of this temperature, because the rise in temperature is small, will make large discrepancies in the calculated weight of condensed steam.

Steam Consumption of Auxiliaries by Calculation of Heat Balance. The heat balance method of the preceding paragraph can be adapted also to the calculation of the steam used by non-condensing auxiliaries discharging their exhaust into a feed-water heater, provided of course all the steam entering the heater is condensed.

w = weight of condensed steam from turbine, lbs.
w_0 = weight of make-up water, lbs.
w_a = weight of steam used by auxiliaries, lbs.
t_1 = temperature of water entering heater, deg. F.
t_2 = temperature of water leaving heater, deg. F.
t_0 = temperature of make-up water, deg. F.
t_a = temperature of steam corresponding to back-pressure, deg. F.
r_a = heat of vaporization, corresponding to back-pressure, B.t.u. per lb.

$$W (t_2 - t_1) + w_0 (t_2 - t_0) = w_a (r_a + t_a - t_2). \quad . \quad . \quad (103)$$

Stage pressures should be observed and recorded when tests are made of steam turbines having a series of pressure stages. These data are often extremely useful, both for checking the weight of condensed steam if the turbine is of the nozzle type[1] and also for showing any abnormal conditions in the several stages.

Results calculated on a basis of kilowatts output should be net; that is, the power required for excitation should be substracted from

[1] Usually the nozzles discharging the steam into the second stage of the turbine are always open, so that the total area is always constant. If, therefore, the areas of the smallest sections of these nozzles are measured and the pressure is observed in the first stage, the weight can be calculated with a considerable degree of accuracy by using the formula for the flow of steam on page 189.

the generator output. If, however, the generator is self-exciting the net output can be measured directly at the terminals of the machine.

"**Guarantee**" **tests** of steam engines and turbines should be made under conditions as nearly as possible the same as that for which the turbine was designed. Different machines will have different correction factors for varying conditions of pressure, superheat, vacuum, etc., so that water rates corrected for large variations are always likely to be more or less inaccurate. This is particularly true in respect to vacuum corrections. Some turbines will give a very good efficiency with a low vacuum, but at a high vacuum because of an insufficiently large steam space the efficiency will be low.

Heat Unit Basis of Efficiency

A **thermal efficiency** can be calculated readily by determining what percentage the heat equivalent of the work is of the heat "used by the turbine," assumed to be the difference between the total heat in the steam at the initial conditions and the heat of the liquid in the condensed steam at the temperature of the exhaust.

By this method the full-load test of a Westinghouse-Parsons turbine reported by F. P. Sheldon & Co., will be calculated from the data given in an official report.

In order to make the results of such calculations of steam turbine tests comparable with the usual heat unit computations of reciprocating steam engine tests the results are often expressed in terms of indicated or "**internal**" **horse power.** It was assumed the mechanical efficiency of a reciprocating engine of about the same capacity at this load was about 93.3 per cent.

THERMAL EFFICIENCY OF A 400-KILOWATT STEAM TURBINE

Brake horse power	660
Corresponding indicated or "internal" horse power of a reciprocating engine $= \frac{660}{.933}$	708
Total steam used per hour, pounds	9169
Steam used per "internal" horse power per hour, pounds	12.96
Steam used per "internal" horse power per minute, pounds	0.216
Steam pressure, pounds per square inch, absolute	166.9
Superheat, degrees Fahrenheit	2.9
Vacuum, referred to 30 inches barometer, inches	28.04
Temperature of condensed steam, degrees Fahrenheit (at .96 pound per square inch absolute pressure)	100.6
Total heat contents of one pound of dry saturated steam at the initial pressure, B.t.u	1193.9
Heat equivalent of superheat in one pound of steam, B.t.u. (C_p from Fig. 305, page 309)	1.9

Total heat contents of one pound of superheated steam, B.t.u.	1195.8
Heat of liquid in condensed steam, B.t.u	68.6
Heat used in turbine per pound steam, B.t.u	1127.2
Heat used in turbine per "internal" horse power per minute, B.t.u (1127.2 × 0.216)	243.5
Heat equivalent of one horse power per minute, B.t.u. $= \frac{33,000}{778}$	42.42
Thermal efficiency, per cent (42.42 ÷ 243.5)	17.4

CALCULATION OF EFFICIENCY (SHAFT AND BUCKET) OF A STEAM TURBINE GENERATOR COMPARED WITH THE RANKINE CYCLE.

1. "Electrical" kilowatts.
2. R.P.M.
3. Steam per hour (corrected for moisture).
4. Water rate per "electrical" kilowatt, pounds per hour. (3) ÷ (1).
5. I^2R, loss in generator, kilowatts.
6. Rotation loss of generator alone, kilowatts.
7. Rotation loss of wheel and generator, kilowatts.
8. "Shaft" kilowatts. (1) + (5) + (6).
9. "Bucket" kilowatts. (1) + (5) + (7).
10. Water rate per "shaft" kilowatt, pounds per hour.
11. Water rate per "bucket" kilowatt, pounds per hour.
12. Steam-chest pressure, pounds per square inch, absolute.
13. Exhaust pressure, pounds per square inch absolute.
14. Available energy, B.t.u.
15. Theoretical water rate, pounds per kilowatt hour, B.t.u. $= \frac{44200}{\text{Avail. En. (14)}}$.
16. "Shaft" efficiency = (15) ÷ (10).
17. "Bucket" efficiency = (15) ÷ (11).

Notes. — For calculating rotation loss of a new design, stage pressures are of course used.

Steam per hour is usually calculated from the area of the nozzles in the first stage if the governor is *not operating*. For a speed-torque test the *flow of steam* is constant and kw. for determining items (8) and (9) are read from this curve $\left(\text{kw.} = \frac{\text{kw.}}{\text{r.p.m.}} \times \text{r.p.m.}\right)$.

The speed output curve (**Fig. 308, page** 316) is very useful to engineers to determine if a turbine is running at its best speed. If the corresponding curves of steam consumption per kilowatt output (usually called water rate per kilowatt) and **efficiency** curves are calculated according to the above form a great deal of information is obtained about the operation and economy of a turbine. The **torque line** in **Fig. 308** is always drawn straight, just as a "Willans line." A curve of **total** steam consumption is usually a straight line for the normal operating limits of a turbine, but usually becomes curved when a by-pass valve opens on overload, or when the turbine is over its capacity so that the pressures are not normal in the stages.

The torque line shows why a **turbine engine is not adaptable to automobiles.** The starting torque of a small commercial turbine is not large, so that starting would be difficult with a small wheel, and reversing and speed reduction would be as difficult as with a gasoline engine. The reciprocating steam engine as well as the gasoline engine has, therefore, advantages over the steam turbine for this service.

Electrical Output of Turbine Generators. Measurement of Direct Current. Careful engineers will not ordinarily use the instruments on the switchboard of a power station for measuring the electrical output of a generator, because, unless exceptional precautions have been taken to avoid " stray " magnetic fields and the instruments have been calibrated in place under operating conditions with a **sufficient interval of time between observations** of current (amperes) at different loads so that the **shunts of the ammeters will reach a constant temperature** for the particular value of current flowing there may be considerable error in the observations. Switchboard voltmeters are usually satisfactory if they are carefully calibrated; but the shunts of the type of ammeters ordinarily used have approximately only 60 millivolts drop, so that the indicating part of the ammeter must be almost entirely a circuit of copper wire. It is for this reason that such instruments are likely to be affected considerably by varying room temperatures, and with some shunt arrangements they are susceptible to errors, also from variations in the value of the circuit itself. For accurate measurements, it is therefore best to use only the **portable** types of indicating ammeters having shunts of 200 millivolts[1] drop. In these latter instruments the indicating part is made up largely of resistance wires having practically no temperature coefficient. Portable voltmeters are also to be preferred to those on the switchboards.

Unless standard shunts of 200 millivolts drop as provided for good portable ammeters are used the influence of " stray " magnetic fields must be guarded against. When on the other hand switchboard instruments are used, such influences must be investigated and arrangements must be devised so that " stray " fields will not affect the measurements. The influence of very weak magnetic fields can be eliminated from the final results by turning the instruments between successive readings. Observations of current (amperes) made with the switchboard type of instruments are also often in error due to thermo-electric effects producing a small electromotive force sufficient, however, to alter the readings of the millivoltmeter. The error due to this cause can be observed by reading the millivoltmeter at the close of the test immediately after the

[1] This value for the drop in shunts is an arbitrary value selected by a number of makers of electrical instruments because it gives the best compensation of all the temperature errors. See *General Electric Review*, February, 1911.

current has been shut off in the main circuit. It should be, of course, the object of the observer to take this reading before the shunts and leads have cooled appreciably. If there is an error due to this cause there will be a small positive or negative deflection of the needle from the correct zero, which should be applied as a correction to all the observations of current.

Measurement of Alternating Current. The same general precautions outlined above for direct-current instruments must be observed in the use of those for alternating current. Although **steady** magnetic fields are not often a cause of much trouble, it happens often, particularly in the case of large generators, that there are large magnetic fields influencing the measuring instruments, which have the same frequency as that of the current measured. To eliminate the effect of such "stray" fields shielded types of instruments should be used. Only with the most expert handling can accurate results be expected when unshielded instruments are used. For measuring large values of alternating current, instrument transformers are generally used. These should be of the precision type and should be sent to a standardizing laboratory before and after a series of tests to be calibrated, and a certificate of accuracy should be obtained. The transformers should be calibrated at as nearly as possible the values of the current to be measured in the tests.

Whenever it is possible tests of generators should be made with a non-inductive load, water rheostats being usually the most satisfactory apparatus for providing such a load. At least tests should be made under conditions giving a low power factor, so that there can be no error in the readings of the instruments due to phase displacements in the instrument transformers. With a purely non-inductive load the readings of the ammeters and the voltmeters can be used to check the wattmeters. Although the **readings of the wattmeters should be taken as the correct value of the output** the apparent power as indicated by the ammeters and voltmeters should agree with the wattmeter readings within one per cent. If a non-inductive load cannot be secured the switchboard ammeters and voltmeters will be satisfactory for readings to indicate whether or not the load on the circuits is properly balanced. **Watt-hour meters** are not usually satisfactory for the accuracy expected in most tests, and the use of these instruments should be generally avoided. It is only in the case where tests must be made under extremely variable service conditions, where it is difficult to obtain a true average from the readings of indicating instruments, that a watt-hour meter, either for direct or for alternating current, may sometimes give more accurate results than the portable indicating types of instruments. Whenever watt-hour meters are used in tests they should be checked **in place**

for a series of constant loads at the frequency, voltage, etc., which are to be used in the test.

Single-phase indicating instruments are to be preferred for measurements of polyphase current to the standard types of so-called polyphase instruments. The reason for this preference is that the indications of a polyphase instrument are produced by two influences from separate phases of the current in such manner that a correction cannot be applied to obtain true values unless the division of the load is determined by the use of single-phase instruments. Obviously, then, if it is necessary to have single-phase instruments in the separate circuits, it is desirable to have them of the precision type, and polyphase instruments are not needed.

RULES FOR CONDUCTING TESTS OF STEAM TURBINES AND TURBO-GENERATORS. A.S.M.E. CODE OF 1912

Determine the object, take the dimensions, note the physical conditions not only of the turbine but of the entire plant concerned, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given on pages 258 to 263 and prepare for the test accordingly.

The apparatus and instruments required for a simple performance test of a steam turbine or turbo-generator, in which the steam consumption is determined by feedwater measurement, are:

(a) Tanks and platform scales for weighing water (or water meters calibrated in place).
(b) Graduated scales attached to the water glasses of the boilers.
(c) Pressure gages, vacuum gages, and thermometers.
(d) A steam calorimeter.
(e) A barometer.
(f) A tachometer or other speed-measuring apparatus.
(g) A friction brake or dynamometer.
(h) Volt meters, ammeters, wattmeters, and watt-hour meters for the electrical measurements in the case of a turbo-generator.

The determination of the heat and steam consumption of a turbine or turbo-generator should conform to the same methods as those described in the Steam Engine Code, pages 294 to 304.

The steam consumed by steam-driven auxiliaries required for the operation of a turbine should be included in the total steam from which the heat consumption is calculated the same as in the case of the steam engine.

Determine what the operating conditions should be to conform to the object in view and see that they prevail throughout the trial.

The rules pertaining to the subjects Duration, Starting and Stopping, Records, and Calculation of Results, are identically the same as those

given under the respective headings in the Steam Engine Code, pages 294 to 298 with the single exception of the matter relating to indicator diagrams and results computed therefrom; and reference may be made to that code for the directions required in these particulars.

DATA AND RESULTS OF STEAM TURBINE OR TURBO-GENERATOR TESTS

(1) Test of............................turbine located at..................
to determine............................conducted by................
(2) Type of turbine and class of service..
(3) Type of generator, kind of current, etc......................................
(4) Rated power of turbine..
(5) Type of boiler..
(6) Kind and type of auxiliaries (air pumps, circulating pumps, feed pumps, etc.)..
(7) Dimensions of turbine or turbo-generator......................................
(8) Dimensions of boilers..
(9) Dimensions of auxiliaries..
(10) Dimensions of condenser..
(11) Date..
(12) Duration..hrs.

Average Pressures and Temperatures

(13) Steam pipe pressure near throttle, by gage....................lbs. per sq. in.
(13a) Absolute steam pipe pressure near throttle..................lbs. per sq. in.
(14) Steam chest pressure by gage............................lbs. per sq. in.
(14a) Absolute steam chest pressure............................lbs. per sq. in.
(15) Barometric pressure of atmosphere in ins. mercury =...........lbs. per sq. in.
(16) Vacuum in condenser:
(a) In inches of mercury..ins.
(b) Corresponding absolute pressure........................lbs. per sq. in.
(17) Exhaust chamber pressure (absolute)........................lbs. per sq. in.
(17a) Temperature in steam pipe near throttle..........................deg. F.
(17b) Temperature in steam chest....................................deg. F.
(18) Temperature of main supply of feedwater to boilers..................deg. F.
(19) Temperature of additional supplies of feedwater.....................deg. F.
(20) Temperature of injection or circulating water entering condenser.......deg. F.
(21) Temperature of injection or circulating water leaving condenser........deg. F.

Total Quantities

(22) Water fed to boilers from main source of supply.........................lbs.
(23) Water fed from additional supplies......................................lbs.
(24) Total water fed to boilers from all sources..............................lbs.
(25) Moisture in steam or superheating near throttle...........per cent or deg. F.
(26) Factor of correction for quality of steam, dry steam being unity.............
(27) Total dry steam consumed for all purposes................................lbs.

Hourly Quantities

(28) Water fed from main source of supply....................................lbs.
(29) Water fed from additional supplies......................................lbs.

(30) Total water fed to boilers per hour.................................lbs.
(31) Total dry steam consumed per hour..................................lbs.
(32) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant..................................lbs.
(33) Net dry steam consumed per hour..................................lbs.
(34) Dry steam consumed per hour:
(*a*) By turbine..................................lbs.
(*b*) By auxiliaries..................................lbs.
(35) Injection or circulating water supplied condensers per hour..............cu. ft.

Heat Data

(36) Heat units per pound of dry steam, based on temperature of Line 18......B.t.u.
(37) Heat units per pound of dry steam, based on temperature of Line 19......B.t.u.
(38) Heat units consumed per hour, main supply of feed....................B.t.u.
(39) Heat units consumed per hour, additional supplies of feed...............B.t.u.
(40) Total heat units consumed per hour for all purposes....................B.t.u.
(41) Loss of heat per hour due to leakage of plant, drips, etc.................B.t.u.
(42) Heat units consumed per hour:
(*a*) By turbine and auxiliaries..................................B.t.u.
(*b*) By turbine alone..................................B.t.u.
(*c*) By auxiliaries..................................B.t.u.

Electrical Data

(43) Average volts, each phase..................................volts
(44) Average amperes, each phase..................................amperes
(45) Average kilowatts, first meter..................................kw.
(46) Average kilowatts, second meter..................................kw.
(47) Total kilowatt output..................................kw.
(48) Power factor..................................
(49) Output consumed by exciter..................................kw.
(50) Net kilowatt output..................................kw.

Speed

(51) Revolutions per minute..................................
(52) Variation of speed between no load and full load........................r.p.m.
(53) Fluctuation of speed on suddenly changing from full load to no load, measured by the increase in the revolutions due to the change..................r.p.m.

Power

(54) Brake horse power..................................b.h.p.
(55) Electrical horse power..................................e.h.p.

Economy Results

(56) Heat units consumed by turbine and auxiliaries per brake h.p.-hr..........B.t.u.
(57) Dry steam consumed per brake h.p.-hr.:
(*a*) By turbine and auxiliaries..................................lbs.
(*b*) By turbine alone..................................lbs.
(*c*) By auxiliaries..................................lbs.
(58) Dry steam consumed per kw.-hr.:
(*a*) By turbine and auxiliaries..................................lbs.
(*b*) By turbine alone..................................lbs.
(*c*) By auxiliaries..................................lbs.

Efficiency Results

(59) Thermal efficiency ratio per brake horse power........................per cent
(60) Ratio of economy of turbine to that of an ideal turbine working with the Rankine cycle..

Work Done per Heat Unit

(61) Ft.-lbs. of net work per B.t.u. consumed by turbine and auxiliaries (1,980,000 ÷ Line 56)..ft.-lbs.

CHAPTER XIV

METHODS OF CORRECTING STEAM TURBINE AND ENGINE TESTS TO STANDARD CONDITIONS

Standard Conditions for Turbine and Engine Tests. If tests of steam turbines and engines could be always made at some **standard vacuum, superheat and admission pressure,** then turbines and engines of the same size and of the same type could be readily compared, and an engineer could determine without any calculations which of two turbines or engines was more economical for at least these standard conditions. But steam turbines and engines even of the same make are not often designed and operated at any standard conditions, so that a direct comparison of steam consumptions has usually no significance.

It will be shown now how good comparisons of different tests can be made by a little calculation involving the reducing of the results obtained for varying conditions to **assumed standard conditions.** The method given here is that generally used by manufacturers for comparing different tests on the same turbine or engine (a " checking " process) or on different types to determine the relative performance. To illustrate the method by an application, a comparatively simple test will first be discussed.

Practical Example. Corrections for Full-load Tests. The curve in **Fig. 309** shows the steam consumption for varying loads obtained from

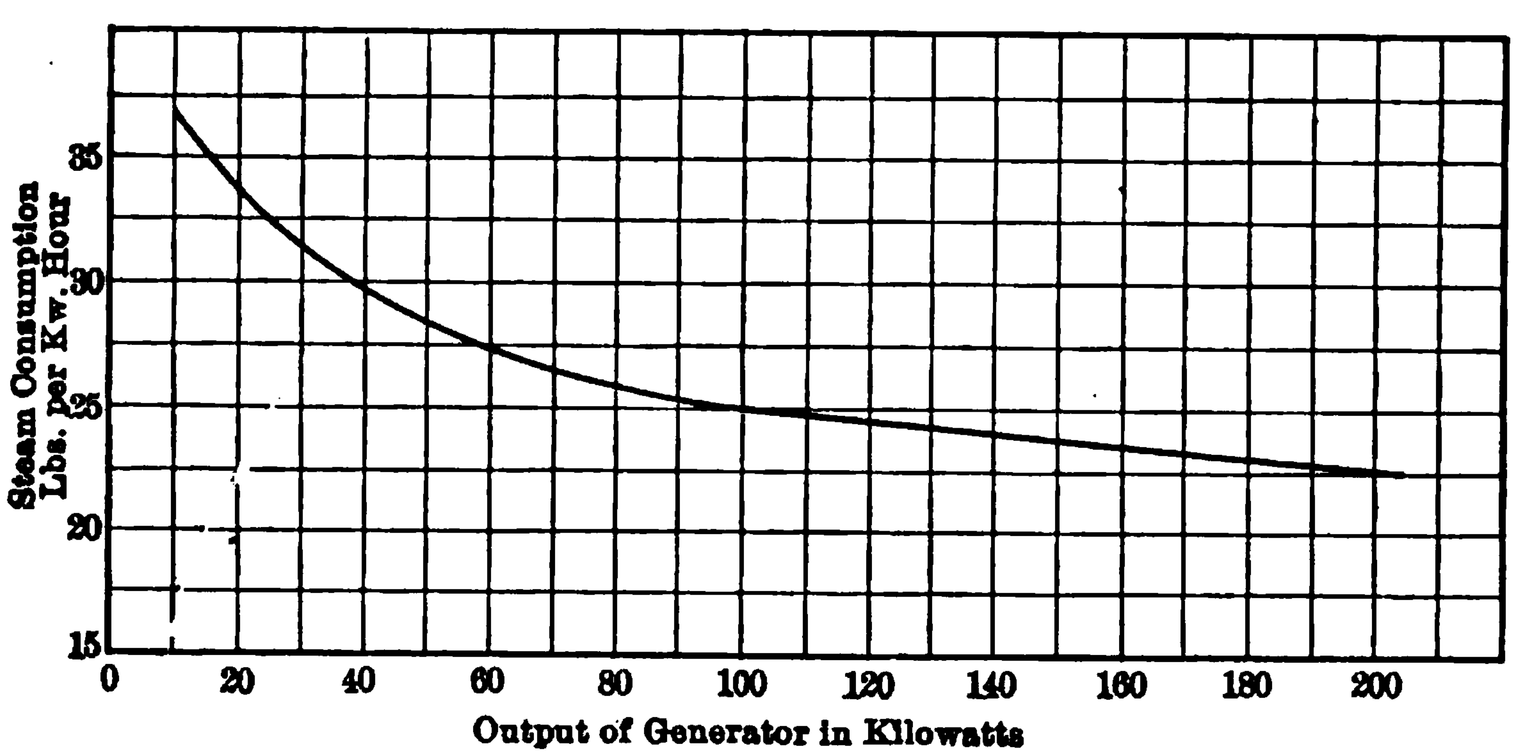


FIG. 309. — Water Rate Curve of a Typical 125-Kilowatt Steam Turbine. (Generator Output.)

tests of a 125-kilowatt **steam turbine** operating at 27.5 inches vacuum, 50 degrees Fahrenheit superheat, and 175 pounds per square inch abso-

lute admission pressure (at the nozzles). It is desired to find the equivalent steam consumption at 28 inches vacuum, 0 degrees Fahrenheit superheat, and 165 pounds per square inch absolute admission pressure for comparison with the "guarantee tests" (**Fig. 310**) of a **steam engine** of about the same capacity operating at the latter conditions of vacuum,

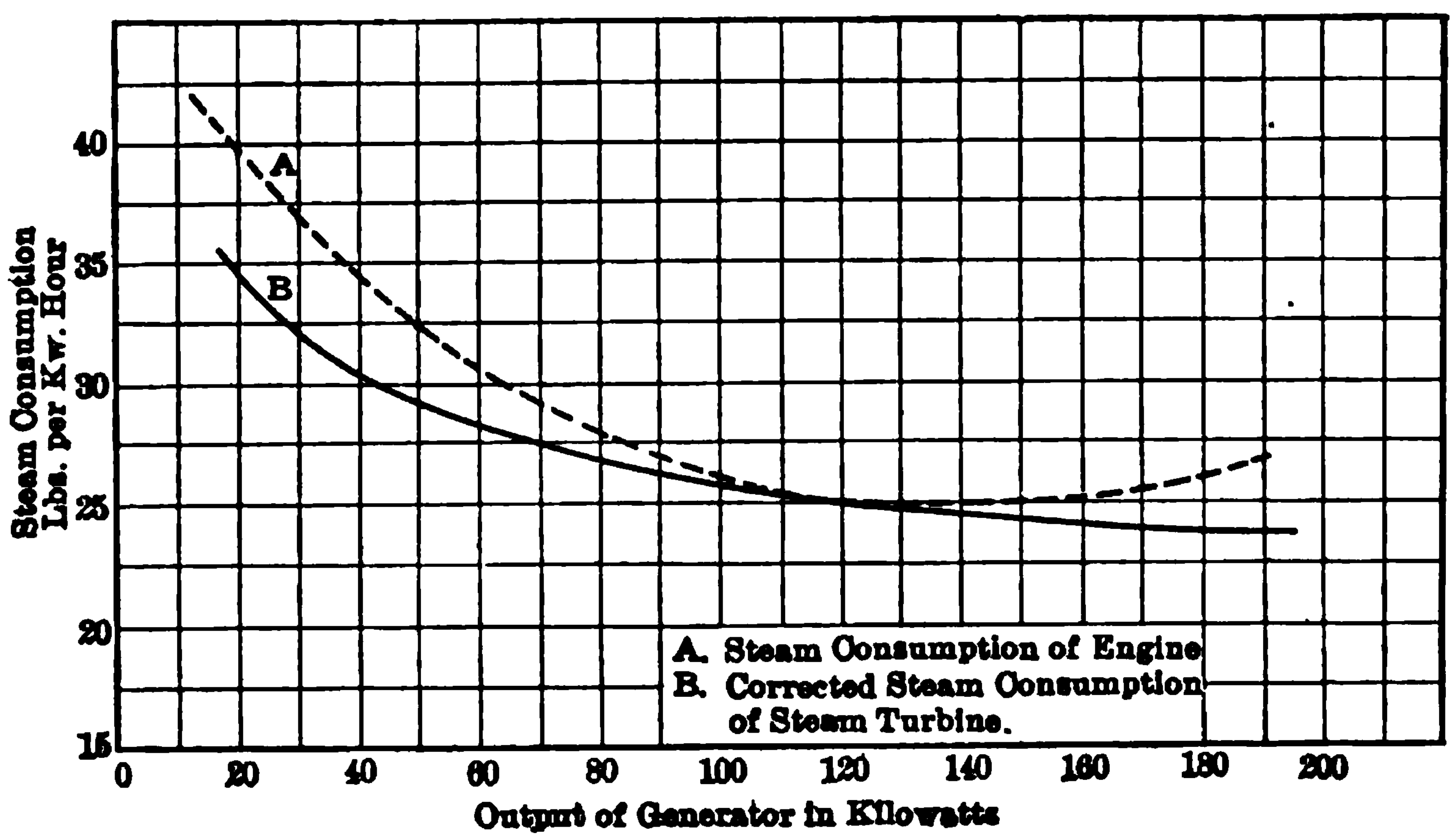


FIG. 310. — Comparative Water Rate Curves of a Reciprocating Steam Engine and a Steam Turbine. (Both with Standard Generators.)

superheat and pressure. The manufacturers of the steam turbine have provided the curves in **Figs. 311, 312, and 313**, showing the change of economy with varying vacuum, superheat and pressure. With the help of these **correction curves,** the steam consumption of the turbine can be

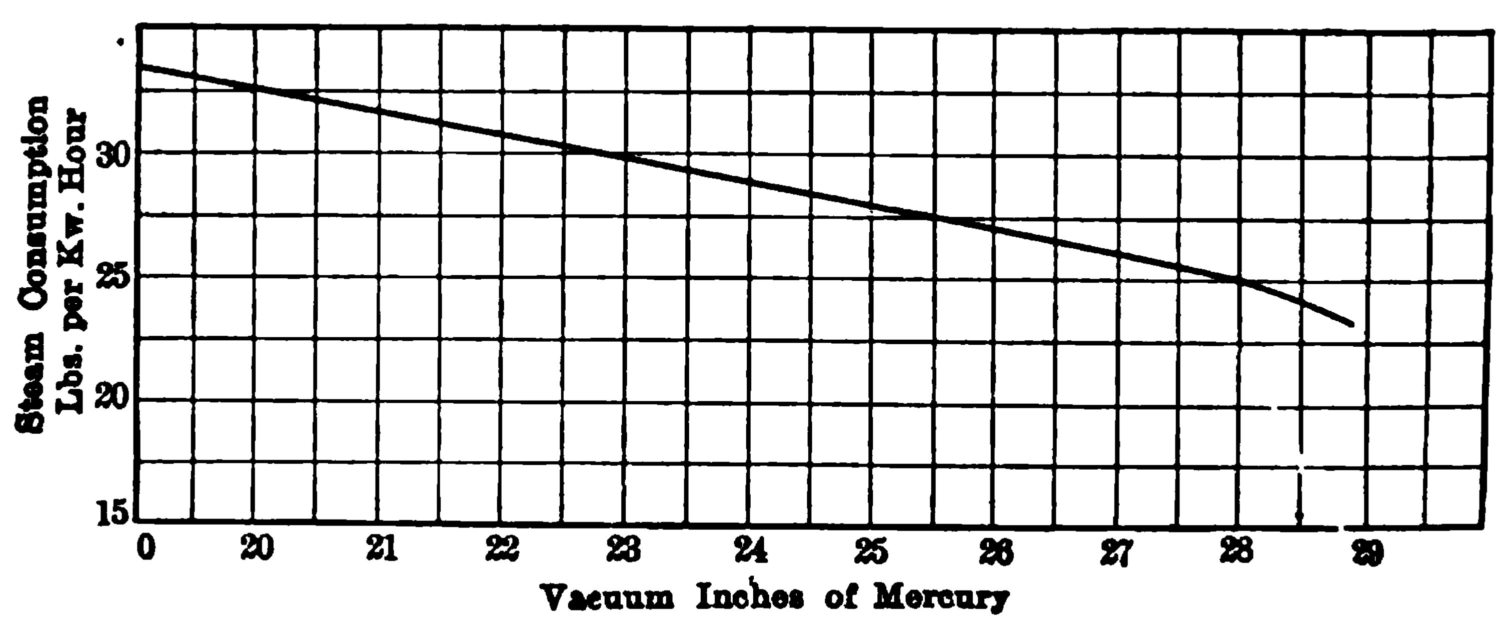


FIG. 311. — Vacuum Correction Curve for a 125-Kilowatt Steam Turbine.

reduced to the conditions of the engine tests. **Fig. 311** shows that between 27 and 28 inches vacuum a difference of 1 inch changes the steam consumption 1.0 pound. **Fig. 312** shows a change of 2.0 pounds per 100 degrees Fahrenheit superheat, and from **Fig. 313** we observe a change of 5.0 pounds in the steam consumption for 100 pounds difference in

admission pressure. Compared with the engine tests the steam turbine was operated at .5 inch lower vacuum, 50 degrees Fahrenheit higher superheat, and 10 pounds higher pressure. At the conditions of the engine tests, then, the steam consumption of the steam turbine should be reduced .5 pound to give the equivalent at 28 inches vacuum, but is increased 1.0 pound to correspond to 0 degrees Fahrenheit superheat, and

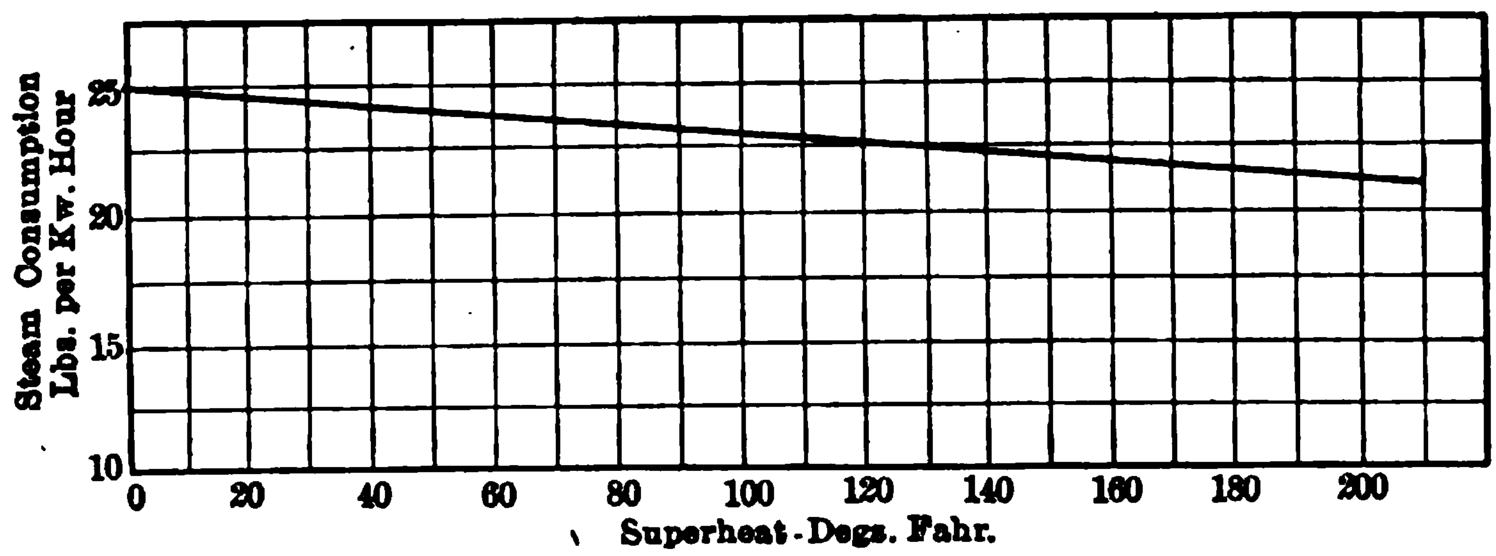


FIG. 312. — Superheat Correction Curve for a 125-Kilowatt Steam Turbine.

.5 pound more to bring it to 165 pounds absolute admission pressure. The full-load steam consumption for the steam turbine at the conditions required for the comparison is, therefore, 24.5 − .5 + 1.0 + .5, or 25.5 pounds.[1]

Persons who are not very familiar with the method of making these corrections will be likely to make mistakes by not knowing whether a

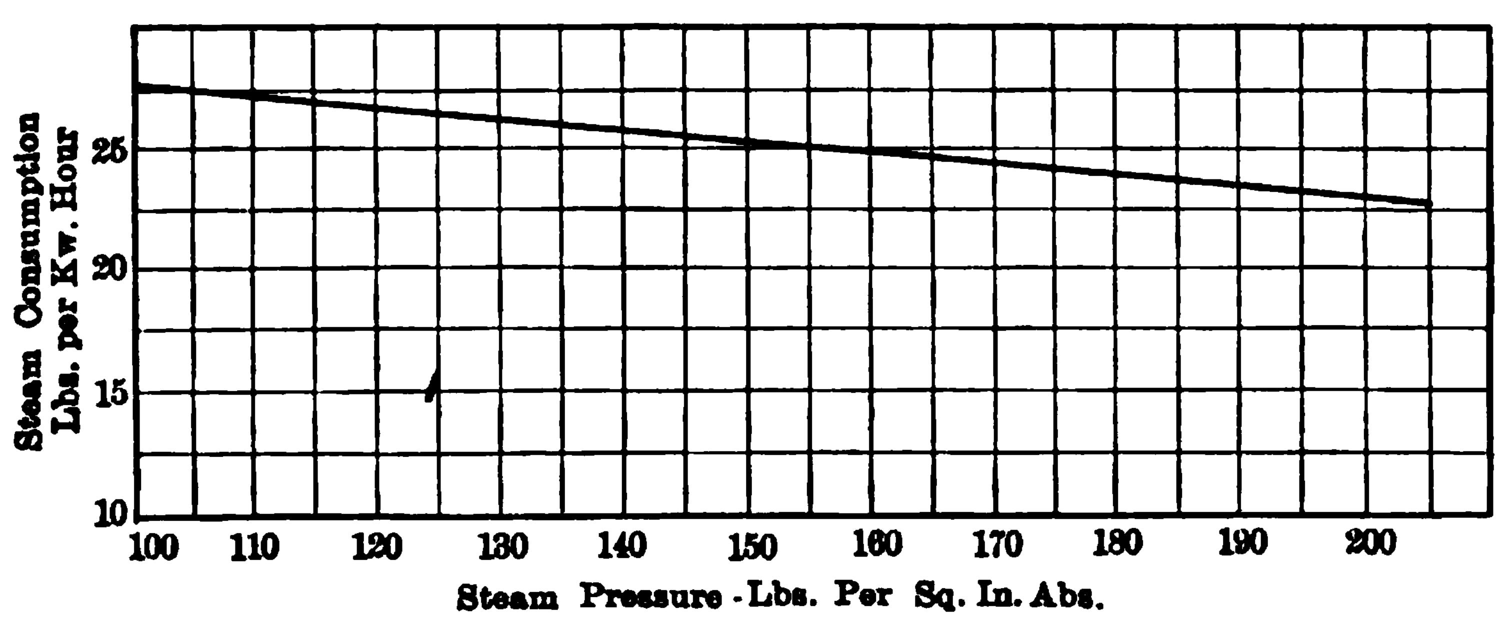


FIG. 313. — Pressure Correction Curve for a 125-Kilowatt Steam Turbine.

correction is to be added or subtracted. A little thinking before writing down the result should, however, prevent such errors. When the per-

[1] The *corrected* steam consumption is found to be nearly the same as that which the three correction curves show for the same conditions, that is, about 25.0 pounds. If there had been a difference of more than about 5 per cent between the corrected steam consumption and that of the correction curves for the same conditions, the "ratio" method as explained on page 332 for fractional loads should have been used also for full load.

formance at a given vacuum is to be **corrected to a condition of higher vacuum, the correction must be subtracted,** because obviously the steam consumption is reduced by operating at a higher vacuum. When the steam consumption with superheated steam is to be determined in its equivalent of dry saturated steam (0 degrees superheat) the correction must be added because with lower superheat there is less heat energy in the steam and consequently there is a larger consumption. Usual corrections for differences in admission pressure are not large; but it is well established that the economy is improved by increasing the pressure.

Corrections for Fractional Loads. It is the general experience of steam turbine manufacturers that **full-load correction curves,** if used by the following "ratio" or percentage method, can be used for correcting **fractional** or **overloads.** This statement applies at least without appreciable error from half to one and a half load, and is the only practicable method for quarter load as well.[1] Stated in a few words, it is assumed then that the steam consumption at fractional loads is changed by the same percentage as at full load for an inch of vacuum, a degree of superheat, or a pound pressure. It will now be shown how this method applies to the correction of the steam consumption of the turbine at fractional loads. Now according to the curve in **Fig. 311,** the steam consumption at 27.5 inches (25.6 pounds) must obviously be multiplied by the ratio[2] $\frac{25.0}{25.6}$, of which the numerator is the steam consumption at 28 inches and the denominator at 27.5 inches, to get the equivalent consumption at 28 inches vacuum. This reasoning establishes the proper method for making corrections; that is, that the **base for the percentage** (denominator of the fraction) must be the steam consumption at the condition to which the correction is to be applied.[3] Similarly the correction ratio to change the consumption at 50 degrees Fahrenheit superheat to 0 degrees Fahrenheit is $\frac{25.0}{24.0}$, and to correct 175 pounds pressure to 165 pounds the ratio is $\frac{24.8}{24.3}$. Data and calculated results obtained by this method may then be tabulated as follows:

[1] A very exhaustive investigation of this has been made by T. Stevens and H. M. Hobart, which is reported in *Engineering,* March 2, 1906.

[2] Assuming that this short length of the curve may be taken for a straight line without appreciable error.

[3] In nearly all books touching this subject so important to the practical, consulting, or sales engineer, the alternative method of taking the steam consumption at the required conditions as the base for the percentage calculation is implied. By such a method percentage correction curves derived from straight lines like Figs. 208 and 209 would be straight lines and, in application, give absurd results. Actually such percentage corrections will fall on curves.

	Conditions of Test.	Required Conditions.	Correction Ratio.	Percentage Correction.
Vacuum, inches	27.5	28	$\frac{25.0}{25.6}$	−2.34%[1]
Superheat, degrees Fahrenheit	50.	0	$\frac{25.0}{24.0}$	+4.17%
Admission pressure, pounds absolute	175.	165	$\frac{24.8}{24.3}$	+2.06%
Net correction	....	....	....	+3.89%

[1] Steps in the calculations are omitted in the table, thus $\frac{25.0}{25.6} = .9766$ or 97.66 per cent making the correction 100 − 97.66 or 2.34 per cent. It may seem unreasonable to the reader that these percentages are calculated to three figures when the third figure of the values of steam consumption is doubtful. In practice, however, the ruling of the curve sheets must be much finer and to larger scale so that the curves can be read more accurately.

The signs + and − are used in the percentage column to indicate whether the correction will increase or decrease the steam consumption. "Net correction" is the algebraic sum of the quantities in the last column.

The following table gives the results of applying the above "net correction" to fractional loads.

	$\frac{1}{4}$ Load 31.3 kw.	$\frac{2}{4}$ Load 62.5 kw.	$\frac{3}{4}$ Load 93.8 kw.	$\frac{4}{4}$ Load 125 kw.	$\frac{5}{4}$ Load 156.3 kw.
Steam consumption from test (Fig. 309)	31.2	26.9	25.2	24.5	23.6
Net correction + 3.89%	+1.2	+1.1	+1.0	+1.0	+0.9
Corrected steam consumption	32.4	28.0	26.2	25.5	24.5

Curve B in **Fig. 310** shows the corrected curve of steam consumption for the steam turbine as plotted from the above table. By thus combining, on the same curve sheet, curves A and B as in this figure, the points of better economy of the turbine are readily understood.

Results of economy tests of various turbines are of very little value for comparison when the steam consumptions or "water rates" are given for all sorts of conditions. With the assistance, however, of curves like those shown in **Figs. 311, 312** and **313**, if they are representative of the type and size of turbine tested, it is possible to make valuable comparisons between two or more different turbines. Some very recent data of Curtis and Westinghouse-Parsons turbines are given below, together with suitable corrections adopted by the manufacturers for similar machines.

The following test of a Westinghouse-Parsons turbine, rated at 7500 kilowatts, was taken at Waterside Station No. 2 of the New York Edison

7500-KILOWATT WESTINGHOUSE-PARSONS TURBINE, WATER-SIDE STATION NO. 2; NEW YORK EDISON COMPANY

		Corrected to	Correction per cent.[1]
Duration of test, hours	8		
Speed revolutions per minute	750		
Average steam pressure, pounds gage	177.5	179	−0.15
Average vacuum, ins. (referred to 30 in. barom.)	27.3	28.5	−3.36
Average superheat, degrees Fahrenheit	95.7	100	−0.29
Average load on generator, kilowatts	9830.5		
Steam consumption, pounds per kilowatt-hour	15.15[2]		
Net correction, per cent			−3.80
Corrected steam consumption, pounds per kilowatt-hour		14.57	

[1] The following corrections were given by the manufacturers and accepted by the purchaser as representative of this type and size of turbine:

Pressure correction .1 per cent for 1 pound.
Vacuum correction 3.5 per cent for 1 inch.
Superheat correction 7.0 per cent for 100 degrees Fahrenheit.
} from *Electric Journal*, November, 1907, page 658.

[2] This is 7½ per cent better than the manufacturer's guarantee.

9000-KILOWATT CURTIS TURBINE, FISK STREET STATION, COMMONWEALTH ELECTRIC COMPANY, CHICAGO

		Corrected to	Correction per cent.[1]
Duration of test			
Speed, revolutions per min	750		
Average steam pressure, pounds gage	179	179	.0
Average vacuum, inches (referred to 30 in. barom.)	29.55	28.5	+12.39
Average superheat, degrees F	116	100	+ 1.28
Average load on generator, kilowatts	8070		
Steam consumption, pounds per kilowatt-hour	13.0		
Net correction, per cent			+13.67
Corrected steam consumption, pounds per kilowatt-hour		14.77	

[1] The following percentage corrections were used:

Superheat corrections 8 per cent for 100 degrees Fahrenheit.
Vacuum correction 8 per cent for 1 inch from curve in Fig. 314.
Pressure correction not given.
} *G. E. Bulletin*, No. 4531.

Co., and a comparison is made with a test of a five-stage 9000-kilowatt Curtis turbine at the Fisk Street Station of the Commonwealth Electric Company of Chicago. As no pressure correction is given for the Curtis machine, the New York Edison test is corrected to the pressure at which the other machine was operated (179 pounds per square inch gage). Approximately an average vacuum for the two tests is taken for the

standard, and 100 degrees Fahrenheit superheat is used for comparing the superheats. These assumed standard conditions make the corrections for each turbine comparatively small. When two tests are to be compared, by far the more intelligent results are obtained if each is corrected to the average conditions of the two tests, rather than correcting one test to the conditions of the other. There is always a chance for various errors when large corrections must be made.

These results show a difference of only .20 pound in the corrected steam consumption, so that for exactly the same conditions these two machines would probably give approximately the same economy. Each turbine is doubtless best for the special conditions for which it was designed.

These results are *equivalent* to respectively 9.58 pounds and 9.72 pounds per **indicated horse power,** assuming 97 per cent as the efficiency of the generator and 91 per cent as the mechanical efficiency of a large Corliss engine according to figures given by Scott.[1]

From experience with other similar turbines it seems as if the vacuum corrections given are too low for each turbine. The correction for the Curtis turbine was obtained from the curve in **Fig. 314,** as given between 27 and 28 inches, while it was used between 28.5 and 29.5 inches, where the curve of steam consumption most likely slopes somewhat as shown by the dotted line in the figure, which was derived from the percentage change of the theoretical steam consumption calculated from the available energy. The correction of 2.7 per cent per inch of vacuum for the Westinghouse-Parsons turbine is probably too low also, although the percentage correction would not be nearly as large as for the Curtis. If both of these corrections are too low, the effect of increasing them would be to increase the corrected steam consumption of the Curtis turbine and reduce that of the Westinghouse-Parsons.

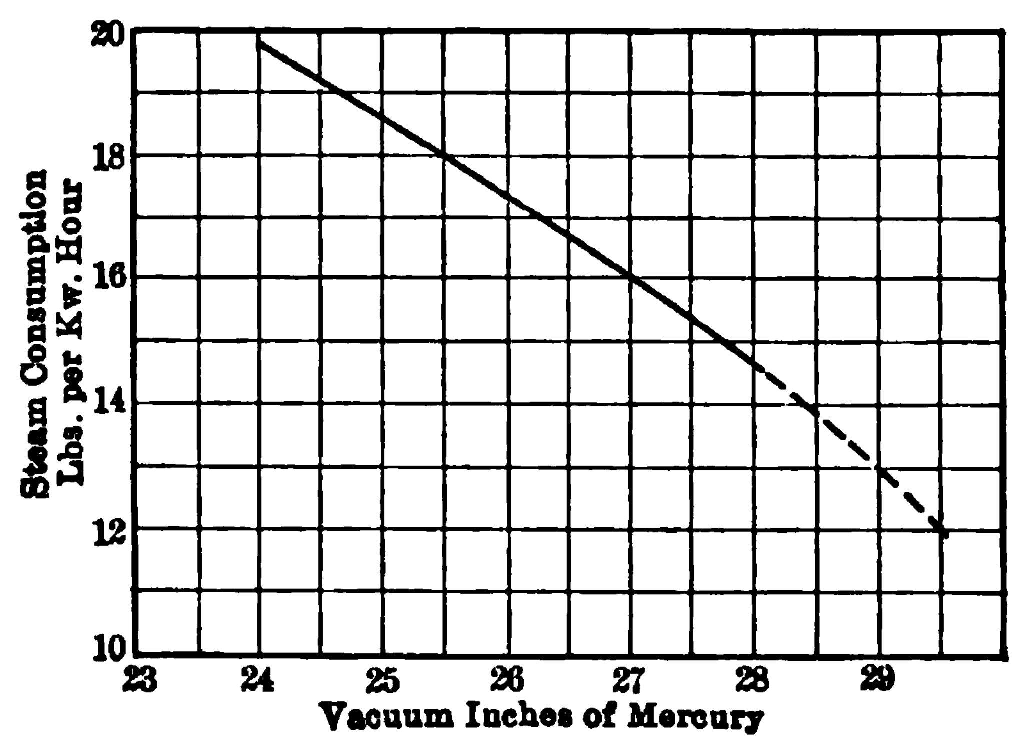


FIG. 314. — Typical Vacuum Correction Curve for a 5000-Kilowatt Impulse Turbine.

[1] *Electric Journal,* July, 1907. It is stated also in this article that the vacuum correction of a Westinghouse-Parsons turbine is 3.5 per cent per inch between 28 and 28.5 inches. Jude states that the vacuum correction for Parsons turbines is 5 to 6 per cent per inch.

CHAPTER XV

TESTS OF COMPLETE STEAM POWER PLANTS

A. S. M. E. CODE OF 1912.

These rules are intended to apply to commercial tests of a complete plant to determine the number of pounds of fuel consumed per unit of work done in a unit of time. For tests of the component parts of a complete plant, such as boilers, engines, turbines, etc., rules may be found in the respective Codes on the preceding pages applying to such cases.

Read the general instructions given on pages 258 to 263. Take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out, and prepare for the test accordingly.

Fuel. Determine the character of the fuel to be used according to the object in view. For further particulars reference may be made to the Boiler Code, page 269.

Apparatus and Instruments. The apparatus and instruments required for a simple performance test of a steam plant are :

(*a*) Platform scales for weighing coal and ashes.
(*b*) Coal calorimeter.
(*c*) Steam engine indicators.
(*d*) A speed-measuring apparatus.
(*e*) Electrical instruments for determining the output of an electric plant.

If the test involves the determination of boiler performance, and engine or turbine performances, additional instruments should be used as pointed out in the respective Codes referring to such tests.

Duration. The duration of a plant test should be not less than one day of 24 hours, and preferably a full week of seven days, including Sunday.

In cases where the engine or turbine is in operation only a part of the day, the duration on which the results are computed should be considered the length of time that the engine or turbine is in operation at its working speed.

Starting and Stopping. In a plant operating continuously, day and night, the times fixed for starting and stopping should follow the regular periods of cleaning the fires. The fires should be quickly cleaned and

then burned low, say to a thickness of 4 inches. When this condition is reached the time should be noted as the starting time, and the thickness of each coal bed observed, as also the water levels and the steam pressure. Fresh coal should then be fired from that weighed for the test, the ashpits thoroughly cleaned, and the regular work of the test proceeded with. At the close of the test, following a regular cleaning, the fires should again be burned low, and when their condition has become the same as that observed at the beginning, the water levels and steam pressure also being the same, the time is observed and this time taken as the stopping time. If the water levels and steam pressure are not the same as at the beginning a suitable correction should be made by computation. The ashes and refuse are then hauled from the ashpits.

In a plant running only a part of the day, and during the remainder of the day the fires are banked, the time selected for the beginning and end of the test should be that following the close of the day's run, when the fires have been burned low preparatory to cleaning and banking. The amount of live coal left on the grates under these circumstances is estimated at the beginning of the test, and the fires brought to the same condition, as near as may be, at the close of the test the next day. If the two quantities differ, a suitable correction is made in the weight of coal fired, as found by calculation.

Records. The general data should be recorded as pointed out on page 296, under the head of Records. Half-hourly readings of the various instruments concerned are usually sufficient, excepting where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 20 minutes, and at more frequent intervals if the nature of the test makes it necessary. Mark on each card the cylinder and the end on which it was taken, also the time of the day. Record on one card of each set the readings of the pressure gages concerned, taken at the same time. These records should subsequently be entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are worked up.

Sampling and Drying Coal. During the progress of the test the coal should be regularly sampled for the purpose of analysis and determination of moisture.

Ashes and Refuse. The ashes and refuse withdrawn from the furnace and ashpit during the progress of the test and at its close should be weighed in a dry state, and, if desired, a representative sample should be obtained for proximate analysis and the determination of the amount of unburned carbon which it contains.

DATA AND RESULTS OF COMPLETE STEAM POWER PLANT TEST

(1) Test of........................plant located at........................
to determine........................conducted by........................
(2) Type of engine or turbine and class of service........................
(3) Rated power of engine or turbine........................
(4) Type of boilers........................
(5) Kind and type of auxiliaries (air pump, circulating pump and feed pump; jackets, heaters, etc.)........................
(6) Dimensions of engine or turbine........................
(7) Dimensions of boilers........................
(8) Dimensions of auxiliaries........................
(9) Dimensions of condenser........................
(10) Date........................
(11) Duration........................hrs.
(12) Length of time engine or turbine was in motion with throttle open..........hrs.
(13) Length of time engine or turbine was running at normal speed............hrs.
(14) Kind of coal........................
(15) Size of coal........................

Average Pressures and Temperatures

(16) Steam pressure at boiler by gage........................lbs. per sq. in.
(17) Steam pipe pressure near throttle, by gage..................lbs. per sq. in.
(18) Barometric pressure of atmosphere in in. of mercury........................
(19) Pressure in receiver by gage........................lbs. per sq. in.
(20) Vacuum in condenser........................ins.
(20a) Temperature of steam near throttle........................deg. F.
(21) Number of degrees of superheating, if any, near throttle..............deg. F.
(22) Temperature of feedwater entering boilers........................deg. F.

Total Quantities, Time, Etc.

(23) Total coal as fired[1]........................lbs.
(24) Moisture in coal........................per cent
(25) Total dry coal consumed........................lbs.
(26) Ash and refuse........................lbs.
(27) Percentage of ash and refuse to dry coal........................per cent
(28) Calorific value by calorimeter test per lb. of dry coal ....B.t.u.
(29) Cost of coal per ton of .... lbs........................dollars

Hourly Quantities

(30) Dry coal consumed per hour, based on duration of running period..........lbs.

Indicator Diagrams

(31) Mean effective pressure in lbs. per sq. in........................

Electrical Data

(32) Total electrical output........................kw.-hr.
(33) Electrical output per hour........................kw.

[1] Where an independent superheater is used, this includes coal burned in the superheater. See also footnote page 277.

(34) Output consumed by exciter...kw.
(35) Net electrical output per hour...kw.
(36) Average volts each phase...volts
(37) Average amperes each phase...amperes
(38) Power factor...

SPEED

(39) Revolutions per minute...

POWER

(40) Indicated horse power developed by main engine:
First cylinder...i.h.p.
Second cylinder...i.h.p.
Whole engine...i.h.p.
(41) Net electrical horse power...e.h.p.

ECONOMY RESULTS

(42) Dry coal consumed per i.h.p. per hour...lbs.
(43) Dry coal consumed per kw.-hr...lbs.
(44) Cost of coal per i.h.p. per hour...cents
(45) Cost of coal per kw.-hr...cents

"Plant" as used in this report should include the entire equipment of the steam plant producing power; that is, the main cylinder or cylinders, the steam jackets and reheaters, the air, circulating and boiler-feed pumps if steam driven, and any other machinery driven by steam required for the operation of the engine. That the engine plant should be charged with the steam used by all the auxiliary machinery in determining the plant economy is necessary because the steam consumption of the engine is finally benefited, or at least it should be, by the heat they return to the system. It is, of course, now the general practice in commercially operated plants to pass the exhaust steam from auxiliaries operated non-condensing through a feedwater heater, thus carrying back to the boiler a great deal of the heat.

When a plant is **operating non-condensing,** discharging the steam into the atmosphere, **or with a jet condenser,** the **steam consumption of the engine cannot be determined by weighing or measuring the steam used** as can be done when a surface condenser is used. The method followed in this case is to determine the steam used by the weight of water supplied to the boiler, assuming, of course, that all the steam from the boiler or boilers used goes to the engine tested. It can usually be arranged for a test of one of the engines in a large plant that one or more boilers can be segregated or cut off in the piping connections so that these boilers alone supply the engine. When, however, this method is to be used it is necessary to determine by a separate test the **leakage of the boiler and of the piping** from the boiler to the throttle valve of the engine. This leak-

age is, of course, the amount of feed-water pumped into the boiler to keep the level in the water gage constant without taking away any more steam than is lost in this way. When determining this leakage, the pressure in the boiler must be maintained the same as that at which steam is to be supplied to the engine for its test. The feed-water pumped into the boiler supplying the engine less the boiler and pipe leakage will be the net amount of steam used by the engine.

CHAPTER XVI

GAS AND OIL ENGINE AND PRODUCER TESTING

The testing of internal combustion engines of the reciprocating type operating with gas, gasoline, kerosene, and alcohol does not differ essentially in the important details from steam engine practice already explained in Chapter XII. Indicator diagrams can be utilized to show the inner workings in the engine cylinder, giving a record of the pressure, " timing " of the valves and ignition for the operation of the engine through a cycle.[1]

Brake horse power is measured with a Prony brake or any other type of dynamometer permitting the determination with facility of the power of the engine. If with a Prony brake or similar device then the **brake horse power** is expressed by the usual formula

$$\text{b.h.p.} = \frac{2\,\pi l n w}{33{,}000}, \quad \ldots \ldots \ldots \quad (104)$$

where l is the length of the brake-arm in **feet**, n is the number of revolutions per minute, and w is the **net** weight indicated by the scales on the brake. Similarly the **indicated horse power** is given by the usual formula for a single-acting steam engine (page 143) except that the **number of explosions** must be used in calculations instead of the number of revolutions, thus,

$$\text{i.h.p.} = \frac{p l a n_e}{33{,}000}, \quad \ldots \ldots \ldots \quad (105)$$

where p is the mean effective pressure in pounds per square inch measured from the indicator diagram, l is the length of the stroke in feet, a is the area of the piston in square inches, and n_e is the number of explosions per minute, then

$$\textbf{Mechanical Efficiency} = \frac{\text{b.h.p.}}{\text{i.h.p.}}. \quad \ldots \ldots \quad (106)$$

Certain important precautions should be observed when taking indicator diagrams, so that a reasonable degree of accuracy may be expected from the results of tests. In the first place the connections

[1] In what is called usually a four-cycle engine there are four piston strokes — one each for suction, compression, expansion, and exhaust, corresponding to two revolutions of the crank shaft for a complete cycle, while in a so-called two-cycle engine, two strokes of the piston make a complete cycle — corresponding to one revolution of the crank-shaft. In the latter case suction and compression are combined in one stroke and expansion and exhaust in another.

between the indicator and the combustion chamber should be as short as possible. It is much more important that in an internal-combustion engine the volume by which the combustion chamber (clearance) is increased by the indicator connections should be small in comparison to the volume of the engine cylinder than in a steam engine; because by increasing the clearance volume, obviously, the pressure resulting from compression is reduced as well as the pressure due to the explosion. It is in this way that large indicators and indicator connections may cause

Fig. 315. — Crosby Gas Engine Indicator.

a considerable reduction in the thermal efficiency of an engine, that is, reducing the efficiency of the transformation of heat energy into work.

Tests of gas engines as made commercially have usually three objects in view:

(1) Brake horse power.

(2) Indicated horse power.

(3) Gas or oil consumption per horse power per hour.

Many types of gasoline engines, particularly those designed for the automobile service, operate at such high speeds that the indicated horse power cannot be obtained with accuracy. In many other kinds of engines classed in this group the mechanical efficiency is very low. It is

for these reasons that such engines are rated by the useful or brake horse power instead of by the indicated horse power, as with steam engines. Brake horse power is therefore the prime criterion by which the performance of these engines is expressed.

Ordinary types of steam engine indicators, moreover, are not very satisfactory for testing gas engines, and many engineers prefer to use one of the type shown in the accompanying illustration, **Fig. 315.** It differs essentially from steam engine indicators of the same type in having in the lower part of the main " barrel," a cylinder of smaller diameter than the one just above it, containing the spring. This smaller cylinder takes a piston of only half the area of the standard size. By this means the scale of the spring ordinarily used is doubled and the shock on the small rods and levers of the pencil motion is only half as great, thus making the liability to breakage and the cost of repairs for such indicators very much less than when a piston of the standard size is used.

Measurement of the Fuel Used. For tests of gas engines the gas used is usually measured by means of a gas meter. A so-called " wet " meter (page 177) is always to be preferred, but if a carefully calibrated dry meter is used very good results can be obtained and it is accurate enough for commercial tests. For gasoline, kerosene and other oil engines the amount of fuel used is preferably determined by direct weighing. The author has found the automatic indicating scales of the **pendulum** type[1] now generally used by grocers and meat dealers to be most satisfactory. By this means the weight of the oil remaining in the " supply " vessel can be observed regularly throughout a test just as with a gas meter, so that irregularities in operation can be immediately observed. The vessel containing the oil used by the engine when placed on a scales must be connected to the carbureter or the pump, as the case may be, with a very flexible metallic tube made without rubber insertion.[2] If an indicating scales is not available a very small platform or grocers' beam scales can be used satisfactorily, although if the weight of fuel oil is desired at regular intervals throughout a test some little time is required to balance the poise.

Another method very commonly applied, however, is to use a cylindrical vessel of small diameter provided with a gage glass in which the level of the liquid can be observed. Such a vessel can be calibrated to determine the weight or volume of oil per inch of height measured on the gage glass.

[1] Very satisfactory scales for this purpose are made by the Toledo Scale Co., of Toledo, Ohio. Similar scales of the **spring** type are not recommended, because necessarily some of the vibrations of the engine will be transmitted to the scales and the indications of the pointer will not be as accurate as they should be.

[2] Tubes of this kind are made by the Pennsylvania Flexible Metallic Tubing Co., Philadelphia.

Observations taken for a test of a gas or oil engine are in general much more uniform than the corresponding data taken in a steam engine trial. For this reason gas engine tests in particular can be made of much shorter duration than upon a steam engine for the same degree of accuracy.

For both gas and oil engines the points plotted on a scale of brake horse power for abscissas and fuel used per unit of time for ordinates will fall along a straight line, similar and resembling the **Willans line** for steam engines and steam turbines (see page 312). A typical set of curves of the results of a test of a gas engine is shown in **Fig. 316.** Brake horse power is taken for the abscissas, as should always be done for gas and oil engine tests and gas used per hour, to its corresponding scale of ordinates, is given by the top curve. Other curves show the gas used per brake horse power per hour and per indicated horse power per hour, the indicated horse power, the **mechanical efficiency**, the **number of explosions** per minute, the **revolutions per minute** and the **thermal efficiency** (heat equivalent of the brake horse power[1] divided by heat supplied).

FIG. 316. — Typical Economy, Speed, Horse Power, and Efficiency Curves of a Five Horse Power Gas Engine.

If the engine is one using gas generated from coal in a producer, the test should cover a long enough period of time to determine with accuracy the coal used in the **gas producer**; such a test should be of at least twenty-four hours' duration, and in **most cases it should preferably extend over several days.**

Gas bags should be placed between the meter and the engine to diminish the variations of pressure, and these should be of a size proportionate to the quantity used. Where a meter is employed to measure the air used by an engine, a receiver with a flexible diaphragm should be placed between the engine and the meter. The temperature and pressure of the gas should be measured, as also the barometric pressure and temperature of the atmosphere.

[1] Most engineers consider the thermal efficiency calculated on the basis of brake horse power more important than that based on indicated power because, particularly for high-speed engines, indicators are not very reliable.

RULES FOR CONDUCTING TESTS OF GAS AND OIL ENGINES. A.S.M.E. CODE OF 1912.

Determine the object, take the dimensions, note the physical condition of the engine and its appurtenances, install the testing appliances, etc., as explained in the general instructions given on pages 258 to 263, and make preparations for the test accordingly.

Apparatus and instruments required for simple performance tests of gas and oil engines are:

(*a*) Tanks and platform scales for weighing oil.
(*b*) A calorimeter for determining the heat of combustion of oil.
(*c*) A gas meter or other apparatus for measuring gas.
(*d*) A gas calorimeter.
(*e*) Pressure gages and thermometers.
(*f*) Gas engine indicators.
(*g*) A planimeter.
(*h*) A speed-measuring apparatus.
(*i*) Gas analyzing apparatus.
(*j*) Scales and tanks for weighing or a water meter for measuring jacket water.
(*k*) A dynamometer.

Duration. The test of a gas or oil engine with substantially constant load should be continued for such time as may be necessary to obtain a number of successive records covering periods of half an hour or less during which the results are found to be uniform. In such cases a duration of from three to five hours is sufficient for all practical purposes.[1]

Starting and Stopping. The engine having been set to work under the prescribed conditions and thoroughly heated (except in cases where the object is to obtain the performance under working conditions), the test is begun at a certain predetermined time by commencing to weigh the oil, or measure the gas, as the case may be, and take other data concerned; after which the regular measurements and observations are carried forward until the end. When the stopping time arrives the test is closed by simply taking the final readings.

Calorific Tests and Analyses. The quality of the oil or gas should be determined by calorific tests and analyses made on representative samples.

CALCULATION OF RESULTS

(*a*) *Heat Consumption.* The number of heat units consumed by the engine is found by multiplying the heat units per lb. of oil or per cu. ft. of gas (higher value), as determined by calorimeter test, by the total weight of oil in lbs. or volume of gas in cu. ft. consumed during the trial.

[1] For tests of maximum power for high-speed engines, it is often impracticable to run tests for a total duration of more than an hour. — Author.

(b) *Horse Power and Efficiency.* The indicated horse power, brake horse power, and efficiency are computed by the same methods as those explained in the Steam Engine Code, on pages 296 to 298, to which reference may be made.

(c) *Heat Balance.* The various quantities showing the distribution of heat in the heat balance given in Table 2, page 348, are computed in the following manner:

The heat converted into work per i.h.p.-hr. (2545 B.t.u.) is found by dividing the work representing 1 h.p., or 1,980,000 ft.-lbs., per hour by the number of ft.-lbs. representing 1 B.t.u., or 778.

The heat rejected in the cooling water is obtained by multiplying the weight of water supplied by the number of degrees rise of temperature, and dividing the product by the indicated horse power.

The heat rejected in the dry exhaust gases per i.h.p.-hr. is found by multiplying the weight of these gases per i.h.p.-hr. by the sensible heat of the gas reckoned from the temperature of the air in the room and by its specific heat. The weight of the dry gases per i.h.p.-hr. may be found by multiplying the weight of dry gas per lb. of carbon, using the formula:

$$\frac{11\,CO_2 + 8\,O + 7\,(CO + N)}{3\,(CO_2 + CO)}, \quad \ldots \ldots \quad (107)$$

in which CO_2, CO, O, and N are expressed in percentages by volume, by the proportion borne by the carbon in the total fuel (either gas or oil), and by the weight of fuel per i.h.p.-hr.

The heat lost in the moisture formed by the burning of hydrogen in the gas is found by multiplying the total heat of 1 lb. of superheated steam at the temperature of the gas, reckoning from the temperature of the air in the room, by the proportion of the hydrogen in the fuel as determined from the analysis, and multiplying the result by 9.

The heat lost through incomplete combustion is obtained by analyzing the exhaust gases and computing the heat of the unburned products which would have been produced by their combustion.

TABLE 1. DATA AND RESULTS OF GAS OR OIL ENGINE TEST—SHORT FORM. CODE OF 1912

(1) Test of........................engine, located at........................
to determine......................conducted by....................
(2) Type and class of engine and number of cycles..............................
(3) Dimensions:
(a) Single or double acting..
(b) Diameter of cylinders..ins.
(c) Stroke of pistons..ft.
(d) Diameter of piston rods..ins.
(e) Compression space or clearance...............................per cent
(f) H.p. constant for 1 lb. m.e.p. and 1 r.p.m.............................
(4) Rated capacity...
(5) Date...
(6) Duration...hrs.
(7) Kind of gas..
(8) Kind of oil..
(9) Physical properties of oil (specific gravity, burning point, and flashing point).....

AVERAGE PRESSURES AND TEMPERATURES

(10) Pressure of gas near meter..ins. water
(11) Barometric pressure......................ins. mercury = ...lbs. per sq. in.
(12) Temperature of cooling water:
(a) Inlet ....deg. F.
(b) Outlet ....deg. F.
(13) Temperature of gas near meter.......................................deg. F.
(14) Temperature of air...deg. F.
(15) Temperature of exhaust gases.......................................deg. F.

TOTAL QUANTITIES

(16) Gas or oil consumed...cu. ft. or lbs.
(17) Cooling water supplied to jackets..lbs.
(18) Calorific value of gas per cu. ft., or of oil per lb., by calorimeter test (higher value)...B.t.u.

HOURLY QUANTITIES

(19) Gas or oil consumed per hour.................................cu. ft. or lbs.
(20) Cooling water supplied to jackets per hour................................lbs.

INDICATOR DIAGRAMS

(21) Maximum pressure...lbs. per sq. in.
(22) Mean effective pressure....................................lbs. per sq. in.

SPEED AND EXPLOSIONS

(23) Revolutions per minute...
(24) Average number of explosions per minute...............................

POWER

(25) Indicated horse power..i.h.p.
(26) Brake horse power..b.h.p.
(27) Friction horse power by difference Line 25 — Line 26..................fr. h.p.
(28) Friction horse power by friction diagrams............................fr. h.p.
(29) Percentage of indicated horse power lost in friction (Line 27)...........per cent

ECONOMY RESULTS

(30) Heat units consumed by engine per hour[1]:
(a) Per indicated horse power.......................................B.t.u.
(b) Per brake horse power...B.t.u.
(31) Pounds of oil or cubic feet of gas consumed per hour:
(a) Per indicated horse power.............................cu. ft. or lbs.
(b) Per brake horse power.................................cu. ft. or lbs.

SAMPLE DIAGRAMS

Note. For an engine driving an electric generator the form may be enlarged to include electrical data in the manner given in the Steam Turbine Code, page 327.

[1] If these results in the case of a gas engine are based on the "lower value" of the heat units, that fact should be so stated.

TABLE 2. DATA AND RESULTS OF GAS OR OIL ENGINE TEST—COMPLETE FORM. CODE OF 1912

(1) Test of........................engine, located at....................
to determine.....................conducted by....................
(2) Type of engine, whether oil or gas..
(3) Class of engine (mill, marine, motor for vehicle, pumping, or other)............
(4) Number of revolutions for one cycle, and class of cycle......................
(5) Method of ignition..
(6) Name of builders..
(7) Dimensions:
(a) Class of cylinder, whether working or compressing..................
(b) Vertical or horizontal..
(c) Single or double acting...
(d) Diameter of cylinders..ins.
(e) Stroke of pistons...ft.
(f) Diameter of piston rods...ins.
(g) Compression space or clearance,..cu. in =..per cent of piston displacement
(h) H.p. constant for 1 lb. m.e.p. and 1 r.p.m..............................
(8) Rated capacity..
(9) Date..
(10) Duration...hrs.
(11) Kind of oil...
(12) Physical properties of oil (specific gravity, burning point, flashing point)........
(13) Kind of gas...

Average Pressures and Temperatures

(14) Pressure of gas near meter.....................................ins. water
(15) Barometric pressure.................ins. mercury =..........lbs. per sq. in.
(16) Temperature of cooling water:
(a) Inlet..deg. F.
(b) Outlet...deg. F.
(17) Temperature of gas near meter...deg. F.
(18) Temperature of air:
(a) By dry-bulb thermometer..deg. F.
(b) By wet-bulb thermometer..deg. F.
(19) Temperature of exhaust gases...deg. F.

Total Quantities

(20) Gas or oil consumed.......................................cu. ft. or lbs.
(21) Air supplied in cu. ft...cu. ft.
(22) Cooling water supplied to jackets...lbs.
(23) Calorific value of oil per lb., or of gas per cu. ft., by calorimeter test (higher value)..B.t.u.

Hourly Quantities

(24) Gas or oil consumed per hour.................................cu. ft. or lbs.
(25) Cooling water supplied per hour..lbs.

Analysis of Oil

(26) Carbon (C) . . . . per cent
(27) Hydrogen (H) . . . . per cent
(28) Oxygen (O) . . . . per cent
(29) Sulphur (S) . . . . per cent
(30) Moisture . . . . per cent

100 per cent

Analysis of Gas by Volume

(31) Carbon dioxide (CO_2) . . . . per cent
(32) Carbon monoxide (CO) . . . . per cent
(33) Oxygen (O) . . . . per cent
(34) Hydrogen (H) . . . . per cent
(35) Marsh gas (CH_4) . . . . per cent
(36) Olefiant gas (C_2H_4) . . . . per cent
(37) Nitrogen (N by difference) . . . . per cent

100 per cent

Indicator Diagrams

(40) Pressure in lb. per sq. in. above atmosphere:
(*a*) Maximum pressure . . . . lbs. per sq. in.
(*b*) Pressure at beginning of stroke . . . . lbs. per sq. in.
(*c*) Pressure at end of expansion . . . . lbs. per sq. in.
(*d*) Exhaust pressure at lowest point . . . . lbs. per sq. in.
(41) Mean effective pressure in lbs. per sq. in . . . .

Speed and Explosions

(42) Revolutions per minute . . . .
(43) Average number of explosions per minute . . . .
(44) Variation of speed between no load and full load . . . . r.p.m.
(45) Fluctuation of speed on suddenly changing from full load to no load measured by the increase in the revolutions due to the change . . . .

Power

(46) Indicated horse power . . . . i.h.p.
(47) Brake horse power . . . . b.h.p.
(48) Friction horse power by difference (Line 46 = Line 47) . . . . fr. h.p.
(49) Friction horse power by friction diagrams . . . . fr. h.p.
(50) Percentage of indicated horse power lost in friction (Line 48) . . . . per cent

Economy Results

(51) Heat units consumed by engine per hour[1]:
(*a*) Per indicated horse power . . . . B.t.u.
(*b*) Per brake horse power . . . . B.t.u.
(52) Pounds of oil or cubic feet of gas consumed per hour:
(*a*) Per indicated horse power . . . . cu. ft. or lbs.
(*b*) Per brake horse power . . . . cu. ft. or lbs.

[1] If these results, in the case of a gas engine, are based on the "LOWER" VALUE OF THE HEAT UNITS, that fact should be so stated. See page 224.

EFFICIENCY

(53) Thermal efficiency ratio:
(a) Per indicated horse power....................................per cent
(b) Per brake horse power....................................per cent

WORK DONE PER HEAT UNIT

(54) Ft.-lbs. of net work per B.t.u. consumed (1,980,000 ÷ Line 51b).........ft.-lbs.

HEAT BALANCE BASED ON B.T.U. PER I.H.P. PER HOUR

	B.t.u.	Per Cent
(55) Heat converted into work..................................	2545	......
(56) Heat rejected in cooling water......................	......	......
(57) Heat rejected in the exhaust gases...........................	......	......
(58) Heat lost due to moisture formed by burning of hydrogen.....	......	......
(59) Heat lost by incomplete combustion.........................	......	......
(60) Heat unaccounted for, including radiation...................	......	......
(61) Total heat consumed per i.h.p. per hour, same as Line 51a....	......	......

SAMPLE DIAGRAMS

Note. For an engine driving an electric generator, the form may be enlarged to include electrical data in the manner given in the Steam Turbine Code, page 327.

Indicator Diagrams of the Suction Stroke of a Gas or Oil Engine. With the ordinary stiff spring used for measuring the horse power of gas and oil engines, very little information regarding the action of the valves during the suction stroke is obtainable from the indicator diagram. For this reason the events in the suction stroke must be obtained with a comparatively light spring, which must be protected, however, from injury when subjected to the excessive pressure of the explosion stroke by inserting a suitable stop of some kind to prevent undue compression of

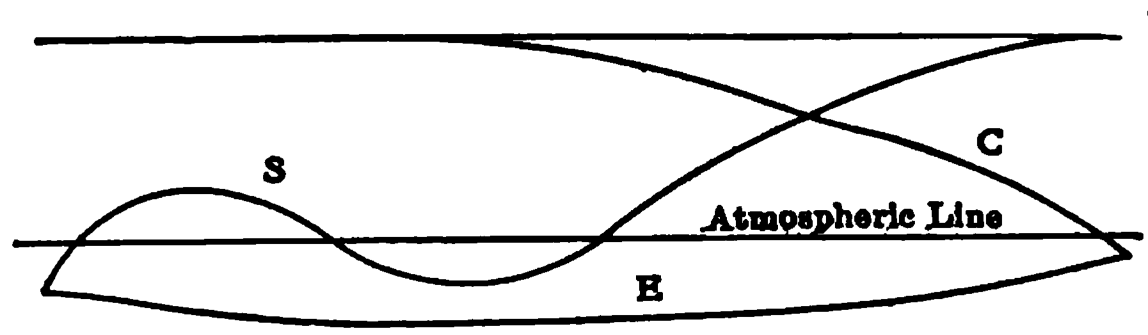


FIG. 317. — "Light Spring" Indicator Diagram of a Gas Engine.

the spring. The device usually adopted is to slip a small brass tube over the piston rod of the indicator to act as a distance piece. Another method, also satisfactory, is to fit a short but very thin brass tube over the outside of the spring, but of such a diameter that it will pass easily inside the cylinder of the indicator and rest freely on the top of the piston. A " light-spring " diagram is shown in **Fig. 317**, which was taken from an engine giving an ordinary diagram like **Fig. 318.** In **Fig. 317** the lower horizontal line is the atmospheric line and the upper horizontal line

is traced by the pencil of the indicator, during the compression and explosion strokes, showing the effect of the stop. The wavy line S shows the exhaust stroke, and the slightly curved line E is the suction. The diagram shows that there was a partial vacuum throughout the suction stroke and for a part of the exhaust stroke, the latter effect being due doubtless to the inertia of the gases in the exhaust pipe.

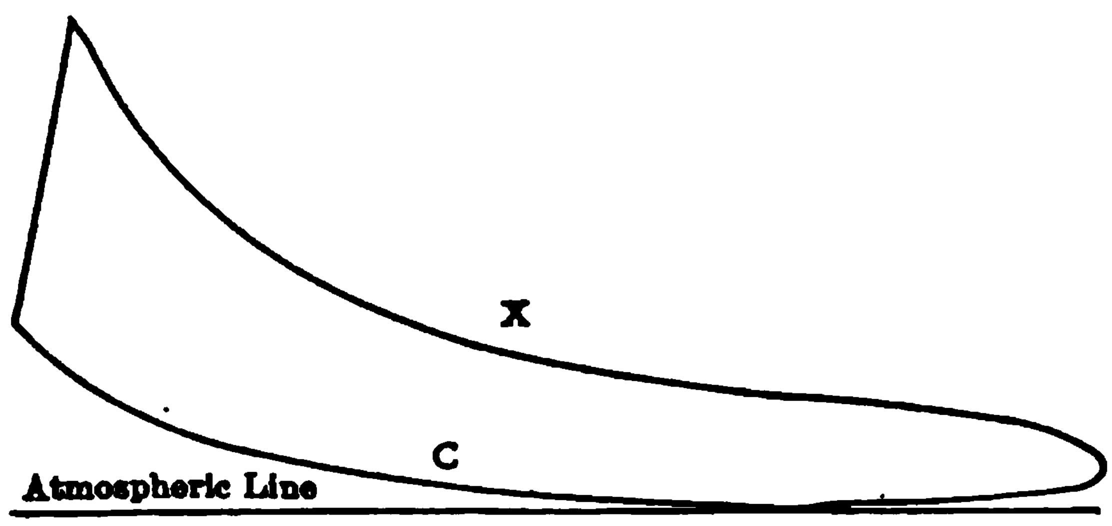


FIG. 318. — Normal Indicator Diagram of a Gas Engine.

In the three figures following very interesting indicator diagrams of gas engines due to Pullen are illustrated. Figs. 319 and 320 show explosions during the suction stroke, generally called explosions in the air pipe, for the reason that since the air valve is then open the explosion occurred probably in the air pipe. In Fig. 319 the effect of the explosion is shown in the indicator diagram by the hump near the atmospheric line near the middle of the stroke, while in Fig. 320 the explosion occurred near the

FIG. 319. — Indicator Diagram of a Gas Engine Showing Explosion in Air Pipe.

end of the stroke. Explosions in the air pipe are sometimes attributed to there being too weak a mixture (too little rich gas) in the cylinder, causing very slow burning instead of a sharp explosion. Under these conditions combustion will not be complete at the end of the working stroke, and this slow burning goes on through the exhaust stroke. Then when the exhaust valve closes, some of this smouldering gas remains in the clearance space, which, when mixed with the new charge during the

next suction stroke, forms a combustible mixture which is easily exploded. Explosions of this nature are generally spoken of as "**back-firing.**" In **Fig. 320** the explosion occurred near the end of the **suction** stroke at **x**, and the air valve has closed before the pressure has had time to fall to atmospheric. On this account the compression line **c** is much higher than it would be under normal conditions as shown by **b**. Since no

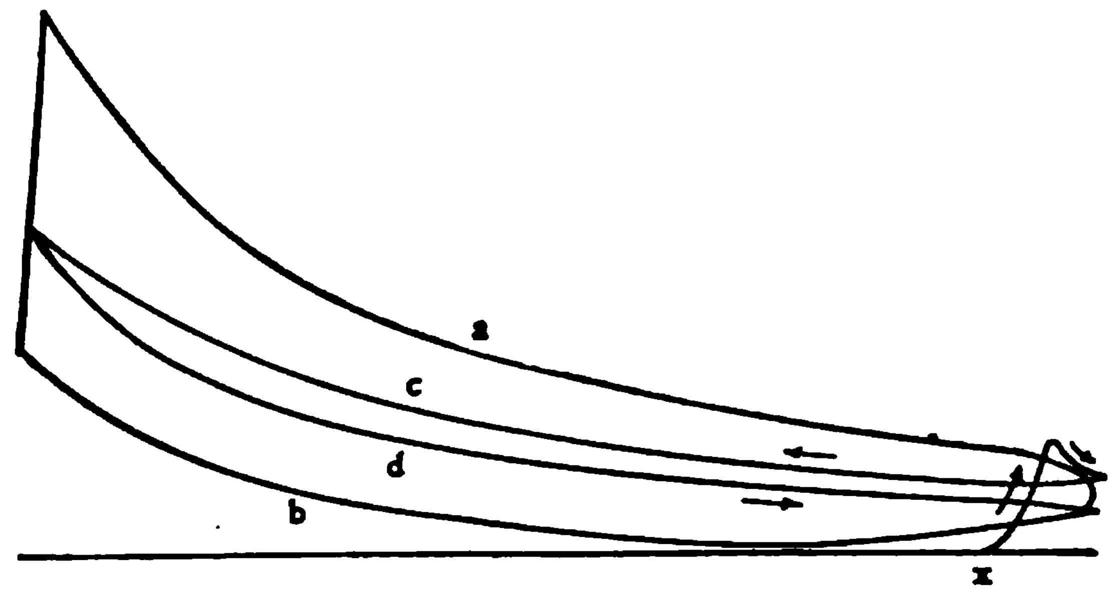


FIG. 320. — Abnormal Gas Engine Diagrams.

explosion takes place at the proper time, the curve **d** corresponding to the working stroke lies just below this abnormal compression line.

No less interesting are the diagrams illustrated in **Fig. 321**, showing the effect of **pre-ignition** on the indicator diagram of a gas engine. Here in two of the diagrams shown the ignition occurred too early or before the end of the compression stroke. Under these conditions there is

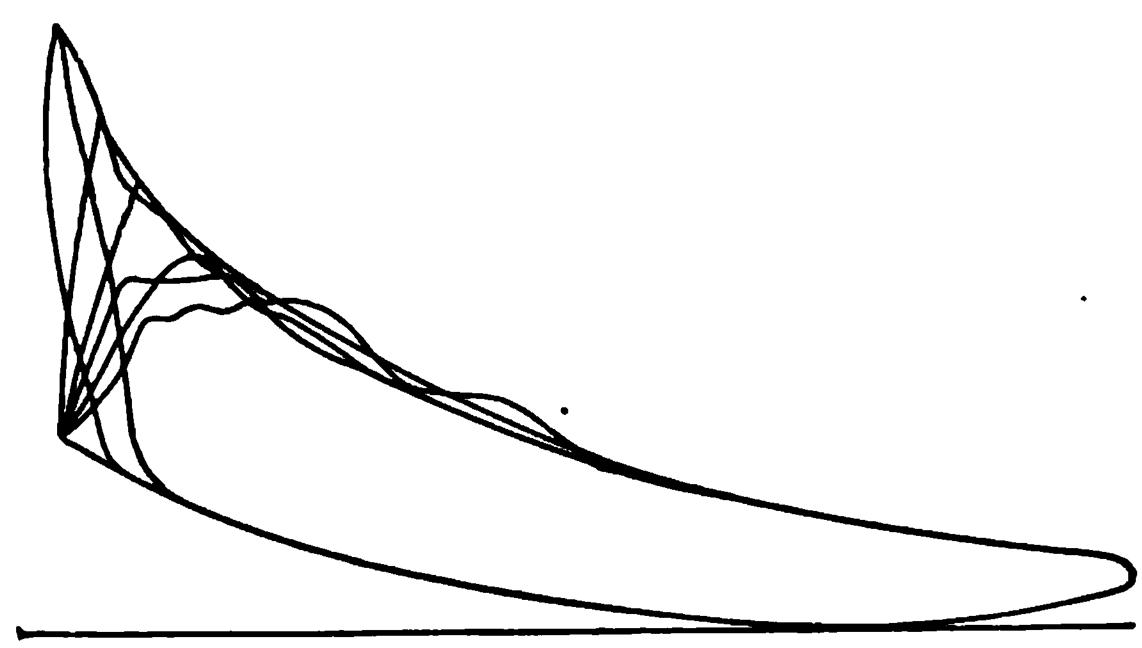


FIG. 321. — Indicator Diagrams of a Gas Engine Illustrating the Effect of "Timing" (from Pre-ignition to Slow Burning).

usually a heavy thumping noise in the engine cylinder, and the engine will not develop as much power as there would be if ignition were "timed" a little later. This effect may be caused in poorly designed engines by some small metal projection or web in the clearance space becoming hot enough to ignite the charge before the ignition device operates. On the other hand, the point of ignition may have been advanced too far by inexperienced persons.

Fuels for Gas and Oil Engines. The ordinary type of gas engine is generally operated with either **illuminating gas** or **natural gas.** Since, however, natural gas occurs only in limited areas its use is very much restricted. **Blast-furnace gas** is used in iron works for operating engines with the waste gases from the blast furnaces. The **gas from coke-ovens,** also a waste gas, is now being used to some extent in the gas engines of power plants in the coke regions. Of the various kinds of so-called fuel gases **producer gas** is for the average engineer by far the most important. Anthracite coal is more easily converted into producer gas than any other fuel, although bituminous coals are now also used. The apparatus used for generating producer gas is called in technical language a **producer.** There are in common use two types of producers for converting solid fuel into a permanent fuel gas. In one type the air or the steam (or both together) that is required for the operation of the producer is forced under pressure, produced usually by a blower, through the bed of solid fuel. In the other type of producer the air and water are drawn through the bed of fuel either by the **suction** of the engine itself, or by the suction of an auxiliary "exhauster." Gas is made at a more or less uniform rate in a **pressure** producer while it operates and the gas is stored in tanks, generally of comparatively small capacity, from which it is drawn to meet the varying needs of the engine. A **suction** producer operating without an auxiliary "exhauster" is not provided with a storage tank, but the gas is generated at the rate demanded by the needs of the engine.

Producer Gas. The most common method for making producer gas to be used in engines is to admit both air and steam (or water vapor) simultaneously and continuously to the incandescent fuel bed. Another method is to burn the fuel for a time with air alone; that is, without any steam, until the fuel bed becomes highly incandescent, and then shut off the supply of air and pass steam or water vapor into the fuel until its temperature becomes so low that very little gas is formed and the air must be used again with the steam supply shut off. The producer continues in operation by alternating the admission of air and steam to the fuel bed. The former of these two methods is the one most generally used.

Suction Gas Producer. Anthracite coal is the most satisfactory fuel for suction gas producers, some preferring "chestnut" size, while others get the most satisfactory operation with the "pea" size if the coal supplied is clean. A producer or generator for such fuel is illustrated in Fig. 322.

Capacity and Efficiency of Gas Producers. The important result to be determined from tests of a gas producer is the ratio of the **heat value of the gas produced** (in B.t.u.) to the **heat value in the same units of the fuel used** and the mechanical or electrical energy required

in producing the gas. The capacity or the rate at which gas can be produced is also important, since a high rate of gasification means lower initial costs of the plant. In reports of tests of gas producers it should be clearly stated whether the higher or the lower heat value of the gas has been used in the calculations. There is no accepted rule as to whether the higher or the lower heat value should be used in guarantees, so that the one to be used must be clearly stated. Probably the best method of stating guarantees is to give the amount (volume) of gas at a standard temperature and pressure and the heat value (higher or lower as preferred) per unit volume (usually a cubic foot) that a producer and its accessories will deliver from a stated weight of coal, of which the heat value per

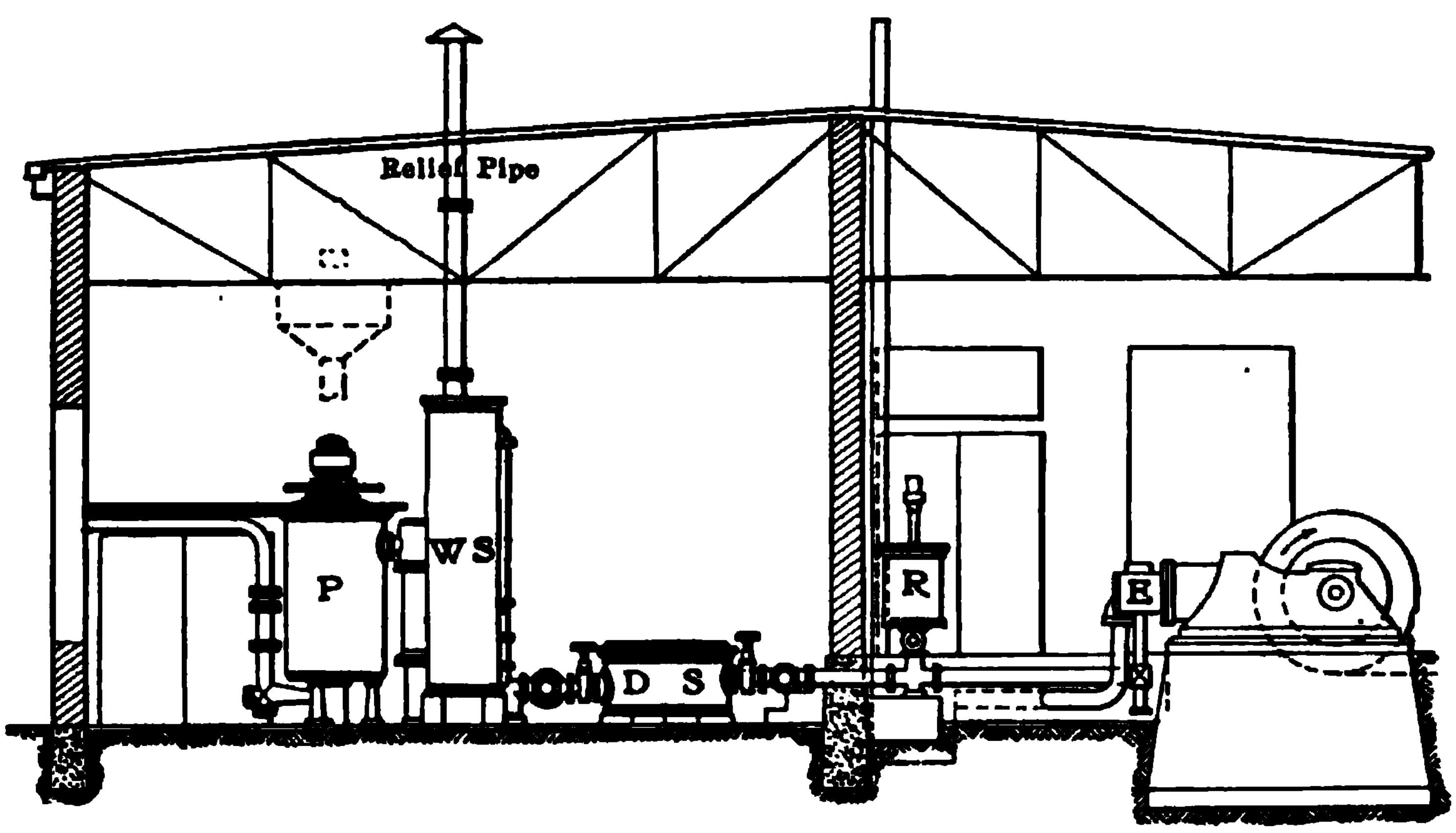


FIG. 322. — Suction Gas Producer and Engine.

pound is also given. Mechanical, electrical, or other energy received from outside sources must also be taken into account. In the specifications for guarantees it should be stated that the loss of unburned fuel in the ash is to be charged as fuel used by the producer.

RULES FOR CONDUCTING TESTS OF GAS PRODUCERS. A.S.M.E. CODE OF 1912

OBJECT AND PREPARATIONS

Determine the object, take the dimensions, and note the physical condition of the producer and its appurtenances.

Fuel.[1] Determine the character of the fuel to be used. If an untried fuel is selected and a test-producer is available, make a preliminary trial of the fuel in this apparatus and ascertain its working characteristics and the proper methods of handling it.

[1] This code is primarily intended for producers using coal. If other fuel, such as wood or oil, is burned, the rules may be modified accordingly.

In tests of maximum efficiency and capacity of a producer for comparison with other producers, the fuel should be some kind of coal which is commercially regarded as a standard for such use in the locality where the test is made. The coal selected for such tests should be the best of its class and free from unusual slag-forming impurities.

Apparatus and Instruments. The apparatus and instruments required for producer tests are:

(a) Platform scales for weighing coal and ashes.
(b) A coal calorimeter.
(c) A gas calorimeter.
(d) Gas analyzing apparatus and appliances for determining tar and soot.
(e) A gas meter, pitot tube, or other suitable apparatus for measuring the gas output.
(f) A manometer or pressure gage.
(g) Water meters for measuring feed and scrubber water, and steam meters for measuring steam used by the apparatus.
(h) Thermometers.

The location of the pitot tube, if used, should be in the delivery pipe at a point near the producer or just beyond the scrubber, or at both points, according to the use made of the gas, either for fuel or power, and other requirements.

Duration. The duration of both efficiency and capacity tests of a producer, with the exceptions noted below, should be such that the total consumption of fuel is at least ten times the weight of the fuel contained in the producer when in normal operation, estimating this weight in the case of coal at 45 lb. per cu. ft.

In cases like down-draft producers which require the fuel bed to be entirely removed and rebuilt at regular intervals, and in producers where a complete cleaning and renewal occurs before the total consumption above stipulated has been reached, the duration should be that of the regular commercial operating cycle, or the time elapsing between two successive renewals of the fuel bed.

Starting and Stopping. The conditions regarding the temperature of the producer and its contents, and the quantity and quality of the latter, should be as nearly as possible the same at the end as at the beginning of the trial. To secure the desired equality of conditions, the starting and stopping should occur at times of regular cleanings, and they should be preceded for a period of not less than 10 hours by the same regular working conditions as those characterizing the test as a whole. The operations of starting and stopping should then be carried on as follows:

(a) *Up-draft Suction Producers.* Remove the ash and clinkers from the grate and the lower part of the furnace space, taking care that the crust or closely-united layer which supports the coal above is not unduly disturbed. Then break open the crust and allow the mass to drop into the space left vacant. Introduce a

poker rod through the poke holes in the upper head and stir up the coal within, thereby causing it to settle and fill the remaining spaces. As a final step, quickly replenish the producer with coal, leaving the hopper level-full, take the time, and consider this the starting time. Then clean the ash pit, and thereafter proceed with the regular work of the test, using weighed coal.

When the time arrives for bringing the trial to a close, the cleaning operations described above are repeated, ending with filling the hopper, taking the time, and considering this the stopping time; finally hauling the ashes and refuse from the ashpit.

(b) *Up-draft Pressure Producers.* Remove the ashes until the top of the ash bed is lowered to the normal working point, say six inches above the blast-hood. Introduce the poker-rod and break down any bridge or crust that may have formed, at the same time closing up the channels that run through the fuel bed, thereby making the bed homogeneous. Then replenish the producer with coal, fill the hopper level-full, take the time, and consider this the starting time. Thereafter proceed with the regular work of the test, using weighed coal.

When the time approaches for closing the test, the operations above described are repeated, ending with replenishing the producer and filling the hopper with weighed coal, taking the time, and considering this the stopping time. The ashes and refuse finally removed are to be dried before weighing, or a sample should be taken and the moisture, as determined therefrom, allowed for.

(c) *Down-draft Pressure Producer.* Thoroughly clean the producer of its entire contents. Introduce a weighed supply of coke or coal, start the fire and build up the fuel bed to its working condition, using weighed coal. When this point is reached, take the time, and consider this the starting time. Thereafter proceed with the regular work of the test.

When the time approaches for closing the test, burn the fuel bed as low as practicable to prepare for cleaning, stop the exhauster, note the time, and consider this the stopping time. Then completely empty the producer, quench the fire remaining in the live coals, separate and weigh the coke and ash, and deduct the weight of the former from that of the coke as charged. Finally dry the ash and refuse, or take a sample and allow for the moisture determined therefrom.

The directions pertaining to Records, Sampling and Drying Coal, Ashes and Refuse, Calorific Tests and Analyses of Coal, are practically the same as those given under the corresponding headings in the Boiler Code, pages 269 and 276.

Calorific Tests and Analyses of Gas Output. The quality of the gas should be determined by calorific tests and analyses, continuous samples for this purpose being taken from the delivery pipe at a point near the producer and at other points as may be needed.

The calorific test should be made with the Junker calorimeter, or its equivalent. Unless otherwise required the "higher value" should be employed in calculating the results of the test. For an approximate determination of the composition of the gas, a modified type of Orsat apparatus may be used, and for complete determinations, the Hempel apparatus or its equivalent. The frequency with which these determinations should be made depends on the uniformity of the output, but the inter-

vals, where practicable, should not be more than one-half hour, the time taken for collecting each sample being not less than one-half hour.

CALCULATION OF RESULTS

(a) *Total Volume of Gas Delivered.* The volume of gas (cu. ft.) found by pitot tube measurement is determined by multiplying the area of the delivery pipe in sq. ft. at the tube by the velocity of the gas in ft. per minute, and the product by the duration of the trial in minutes. The equivalent volume at atmospheric pressure (30 in. barometer) and temperature of 62 deg. Fahr. is obtained by the usual method of thermodynamics as explained on page 226.

(b) *Net Volume of Gas Delivered.* The net volume of gas delivered is found by subtracting from the total volume the equivalent volume of gas required for furnishing steam drawn from an outside source, if any, or for furnishing power used for any purpose concerned in the operation of the producer and its auxiliaries.

(c) *Weight of Gas.* The weight of dry gas delivered is found by multiplying the volume in cu. ft., reduced to 62 deg. and 30 in. barometer, given in Table 2, page 360, Line 21, by the weight per cu. ft. of gas given in Table 2, Line 77.

The weight of the gas per cu. ft. is determined by multiplying the percentage of each component gas as found by analysis (see Lines 55 to 63, Table 2, page 361) by its weight in lb. per cu. ft., at 62 deg. and 30 in., barometer as given in the following table, and dividing the sum of the products by 100.

CO_2	0.1116	CH_4	0.0428
CO	0.0736	C_2H_4	0.0737
O_2	0.0842	SO_2	0.1638
H_2	0.0053	H_2S	0.0868
N_2	0.0740		

(d) *Moisture in Gas Leaving Producer.* The percentage of moisture in the gas is found by passing a measured sample of the gas through a chloride of calcium tube and weighing the amount of moisture absorbed.

(e) *Percentage of Tar and Soot in Gas.* The percentage of tar and soot is found by comparing the total weight determined, including that collected from the various tar drips with the total weight of dry fuel used.

(f) *Efficiency.* The efficiency is the relation between the calorific value of the gas per lb. of fuel charged, or combustible burned, and the calorific value of 1 lb. of fuel or combustible. The former is ascertained by multiplying the B.t.u. per cu. ft. of gas as determined by the calorimeter test (higher value) by the cu. ft. of gas delivered, and dividing the product by the total weight of fuel charged or combustible burned.

The "combustible burned" is determined by subtracting from the weight of coal charged the moisture in the coal and the ash and refuse, including unburned coal which is withdrawn from the producer or ash-pit during the progress of the trial. The "combustible" used for determining the calorific value is the weight of the coal less the moisture and ash found by analysis.

The efficiency of "conversion and cleaning" in the above calculation is found by using the total volume of gas delivered. The "efficiency of the plant" is found by using the net volume of gas delivered.

(g) *Heat Balance.* The various quantities showing the distribution of heat in the heat balance given in Table 2, page 361, are computed in the following manner:

The heat contained in the dry gas is found by multiplying the cubic feet of gas at 62 deg. and 30 in. barometer per lb. of dry coal by the calorific value of 1 cu. ft. of gas at 62 deg. and 30 in. barometer (higher value).

The heat carried away by the scrubber is obtained by multiplying the weight of water fed to the scrubber by the number of degrees rise of temperature, and dividing the product by the total weight of dry coal consumed.

The heat contained in the moisture leaving the producer is found by multiplying the total weight of dry gas by the proportion of moisture in the gas leaving the producer and by the total heat of 1 lb. of superheated steam at the temperature of the gas leaving the producer reckoned from the temperature of the air in the room, and dividing the product by the weight of dry coal consumed.

Chart. In trials having for an object the determination and exposition of the complete performance from beginning to end, the entire log of readings and data should be plotted on a chart and represented graphically. (See **Fig. 296**, page 268.)

TABLE 1. DATA AND RESULTS OF GAS PRODUCER TEST — SHORT FORM. CODE OF 1912

(1) Test of..........................producer located at.....................
to determine.......................conducted by...................
(2) Type of producer...
(3) Rated capacity of producer...................................
(4) Date..
(5) Duration...hrs.
(6) Kind of coal[1] and where mined...............................
(7) Size of coal..

AVERAGE PRESSURES, TEMPERATURES, ETC.

(8) Steam pressure in vaporizer by gage.........................lbs. per sq. in.
(9) Gas pressure in delivery main at point where gas is measured.........ins. water
(10) Temperature of feedwater...................................deg. F.
(11) Temperature of gas in delivery main near producer..................deg. F.
(12) Temperature of gas in delivery main at point where gas is measured.....deg. F.
(13) Force of blast or draft in ashpit.............................ins. water

TOTAL QUANTITIES

(14) Weight of coal as charged.....................................lbs.
(15) Percentage of moisture in coal.................................per cent
(16) Total weight of dry coal consumed..............................lbs.
(17) Total ash and refuse...lbs.
(18) Percentage of ash and refuse in dry coal.......................per cent
(19) Total cu. ft. of gas as measured...............................cu. ft.
(20) Equivalent cu. ft. of gas at 62 deg. F. and 30 in. barometer..............cu. ft.
(21) Net cu. ft. of gas at 62 deg. F. and 30 in.[2] barometer ....cu. ft.
(22) Total water fed to vaporizer...................................lbs.
(23) Total water supplied to scrubber...............................lbs.

[1] If other fuel than coal is used the lines may be changed to read accordingly.
[2] After deducting equivalent gas required for auxiliaries.

Hourly Quantities

(24) Dry coal consumed per hour.......................................lbs.
(25) Dry coal per sq. ft. of main fuel bed per hour...........................lbs.
(26) Total cu. ft. of gas delivered per hour..............................cu. ft.
(27) Total cu. ft. of gas per hour at 62 deg. F. and 30 in. barometer..........cu. ft.
(28) Net cu. ft. of gas per hour at 62 deg. F. and 30 in. barometer.............cu. ft.

Economy Results

(29) Total cu. ft. of gas delivered per lb. of dry coal (Line 13 ÷ Line 10).......cu. ft.
(30) Equivalent total gas at 62 deg. F. and 30 in. barometer per lb. of dry coal..cu. ft.
(31) Net cu. ft. of gas at 62 deg. F. and 30 in. barometer per lb. of dry coal[1]....cu. ft.
(32) Net cu. ft. of gas at 62 deg. F. and 30 in. barometer per lb. of combustible.cu. ft.

Efficiency[2]

(33) Calorific value of dry coal per lb.................................B.t.u.
(34) Calorific value of combustible per lb..............................B.t.u.
(35) Calorific value of gas per cu. ft. (higher value).........................B.t.u.
(36) Efficiency of producer based on coal..............................per cent
(37) Efficiency of producer based on combustible.........................per cent

Cost of Production

(38) Cost of coal per ton of .... lbs. delivered...........................dollars
(39) Cost of coal required for producing 10,000 net cu. ft. of gas at 62 deg. F. and 30 in. barometer...dollars
(40) Cost of coal required for producing one million B.t.u. in the gas.........dollars

TABLE 2. DATA AND RESULTS OF GAS PRODUCER TEST—COMPLETE FORM. CODE OF 1912

(1) Test of....................producer located at..........................
to determine...................conducted by.........................

Dimensions

(2) Outside diameter of producer...ft.
(3) Height of producer..ft.
(4) Inside diameter of producer..ft.
(5) Diameter of grate...ft.
(6) Area of grate...sq. ft.
(7) Percentage of air space in grate...................................per cent
(8) Area of fuel bed (at maximum diameter)............................sq. ft.
(9) Area of water-heating surface in vaporizer...........................sq. ft.
(10) Rated capacity of producer in lbs. of coal per hour.....................lbs.
(11) Date...
(12) Duration ....hrs.
(13) Kind of coal[3] and where mined...
(14) Size of coal...

[1] After deducting equivalent gas required for auxiliaries.

[2] If the efficiency is based on the "lower value" of the heat units in the gas the fact should be so stated.

[3] If other fuel than coal is used the lines may be changed to read accordingly.

Average Pressures, Temperatures, Etc.

(15) Steam pressure in vaporizer by gage..........................lbs. per sq. in.
(16) Gas pressure in main at point where gas is measured................ins. water
(17) Force of blast or draft in ashpit.................................ins. water
(18) Barometric pressure...ins. mercury
(19) Temperature of feedwater entering vaporizer..........................deg. F.
(20) Temperature of gas in main near producer.............................deg. F.
(21) Temperature of gas in main at point where gas is measured............deg. F.
(22) Temperature of air in room...deg. F.
(23) Temperature of water entering scrubber...............................deg. F.
(24) Temperature of water leaving scrubber................................deg. F.
(25) Weight of dry gas per cu. ft. reduced to 62 deg. F. and 30 in. barometer......lbs.

Total Quantities

(26) Weight of coal as fired..lbs.
(27) Percentage of moisture in coal......................................per cent
(28) Total weight of dry coal consumed...lbs.
(29) Total ash and refuse...lbs.
(30) Percentage of ash and refuse in dry coal............................per cent
(31) Total number of cu. ft. of gas as measured...........................cu. ft.
(32) Equivalent cu. ft. of gas at temperature of 62 deg. F. and pressure of atmosphere of 30 in. barometer..cu. ft.
(33) Net cu. ft. of gas at 62 deg. F. and 30 in.[1] barometer.................cu. ft.
(34) Total weight of dry gas..lbs.
(35) Percentage of moisture in gas leaving producer......................per cent
(36) Percentage of tar and soot in gas referred to total fuel................per cent
(37) Total water fed to vaporizer..lbs.
(38) Total water evaporated in vaporizer.......................................lbs.
(39) Total weight of steam supplied to producer................................lbs.
(40) Total weight of water fed to scrubber.....................................lbs.

Hourly Quantities

(41) Dry coal consumed per hour...lbs.
(42) Dry coal consumed per hour per sq. ft. of grate...........................lbs.
(43) Dry coal consumed per hour per sq. ft. of main fuel bed...................lbs.
(44) Total cu. ft. of gas delivered per hour (Line 20 ÷ Line 17)..............cu. ft.
(45) Total cu. ft. of gas per hour at 62 deg. F. and 30 in. barometer..........cu. ft.
(46) Net cu. ft. of gas delivered per hour at 62 deg. F. and 30 in. barometer....cu. ft.
(47) Weight of dry gas per hour..lbs.
(48) Water fed per hour to vaporizer...lbs.
(49) Water evaporated per hour in vaporizer....................................lbs.
(50) Steam supplied to producer per hour.......................................lbs.
(51) Water fed to scrubber per hour..lbs.

Proximate Analysis of Coal

(52) Fixed carbon...per cent
(53) Volatile matter..per cent

[1] After deducting equivalent gas required for auxiliaries.

(54) Moisture..per cent
(55) Ash...per cent

100 per cent

(56) Sulphur, separately determined...................................per cent

ULTIMATE ANALYSIS OF DRY COAL

(57) Carbon (C)..per cent
(58) Hydrogen (H_2)..per cent
(59) Oxygen (O_2)..per cent
(60) Nitrogen (N_2)..per cent
(61) Sulphur (S)...per cent
(62) Ash..per cent

100 per cent

(63) Moisture in sample of coal as received...........................per cent

ANALYSIS OF ASH AND REFUSE

(64) Carbon...per cent
(65) Earthy matter..per cent

ANALYSIS OF GAS BY VOLUME[1]

(66) Carbon dioxide (CO_2)..per cent
(67) Carbon monoxide (CO)...per cent
(68) Oxygen (O_2)...per cent
(69) Hydrogen (H_2)...per cent
(70) Marsh gas (CH_2)...per cent
(71) Olefiant gas (C_2H_4)..per cent
(72) Sulphur dioxide (SO_2)...per cent
(73) Hydrogen sulphide (H_2S).......................................per cent
(74) Nitrogen (N_2 by difference)...................................per cent

100 per cent

CALORIFIC VALUES BY CALORIMETER

(75) Calorific value of dry coal per lb...................................B.t.u.
(76) Calorific value of combustible per lb................................B.t.u.
(77) Calorific value of gas per cu. ft. at 62 deg. F. and 30 in. barometer (higher value)...B.t.u.

ECONOMY RESULTS

(78) Total cu. ft. of gas as measured, per pound of dry coal consumed.........cu. ft.
(79) Equivalent cu. ft. of gas at 62 deg. F. and 30 in. barometer per lb. of dry coal. cu. ft.
(80) Equivalent cu. ft. of gas at 62 deg. F. and 30 in. barometer per lb. of combustible...cu. ft.
(81) Net cu. ft. of gas at 62 deg. F. and 30 in. barometer per lb. of dry coal....cu. ft.
(82) Net cu. ft. of gas at 62 deg. F. and 30 in. barometer per lb. of combustible..cu. ft.

[1] Sample of gases should be collected at producer outlet before the gases pass through the scrubber as some of the carbon dioxide and hydrocarbons are absorbed in the scrubbers.

EFFICIENCY[1]

(83) Efficiency of producer based on coal:
(a) Conversion and cleaning.......per cent
(b) Plant.......per cent
(84) Efficiency of producer based on combustible:
(a) Conversion and cleaning.......per cent
(b) Plant.......per cent

COST OF PRODUCTION

(85) Cost of coal per ton of .... lbs., delivered.......dollars
(86) Cost of coal required for producing 10,000 net cu. ft. of gas at 62 deg. F. and 30 in. barometer.......dollars
(87) Cost of coal for producing 1,000,000 B.t.u.......dollars

HEAT BALANCE BASED ON 1 LB. OF DRY COAL

	B.t.u.	Per cent
(88) Heat contained in dry gas.......	......	......
(89) Heat carried away by scrubber.......	......	......
(90) Heat contained in moisture leaving producer.......	......	......
(91) Heat unaccounted for, including radiation, — difference between the sum of Lines 88, 89, and 90 and Line 92.......	......	......
(92) Total calorific value of 1 lb. of dry coal, same as Line 75......	......	......

RULES FOR CONDUCTING TESTS OF COMPLETE GAS POWER PLANTS. A.S.M.E. CODE OF 1912

OBJECT AND PREPARATIONS

The usual object of testing a complete gas power plant, embracing producer, engine, and appurtenances, is the determination of its commercial performance, i.e., the number of pounds of fuel consumed per unit of work done in a unit of time, and the rules given in this code apply to tests having that object. For directions pertaining to tests of the producer and engine individually, reference may be made to the Producer and Gas Engine Codes, pages 345 and 354. Determine the character of the fuel to be used.

The duration of a gas producer plant test should conform to that of the producer alone, rules pertaining to which may be found in the Gas Producer Code.

In cases where the engine is in operation only a part of the day, the hourly consumption of coal from which the economy results are computed should be the total coal burned in the producer divided by the number of hours that the engine is in operation at its working speed.

The rules for starting and stopping a complete plant test are governed by those required for starting and stopping the test of the producer, which are those given in the Producer Code, page 355.

[1] If the efficiency is based on the "lower value" of the heat units in the gas, the fact should be so stated.

DATA AND RESULTS OF TEST OF COMPLETE GAS POWER PLANT. CODE OF 1912

(1) Test of......................... gas power plant at....................
to determine........................ conducted by..................
(2) Type and dimensions of producers..
(3) Rated capacity of producers..
(4) Total area of main fuel bed at maximum diameter.................... sq. ft.
(5) Type and dimensions of engine..
(6) Rated power of engine...
(7) Date...
(8) Duration.. hrs.
(9) Kind of coal..
(10) Size of coal...

Average Pressures and Temperatures

(11) Pressure of gas near throttle valve........................... ins. water
(12) Barometric pressure.. ins. water
(13) Temperature of cooling water leaving engine...................... deg. F.
(14) Temperature of air in room.. deg. F.

Total Quantities

(15) Weight of coal as charged.. lbs.
(16) Percentage of moisture in coal..................................... per cent
(17) Total weight of dry coal consumed.................................... lbs.
(18) Total ash and refuse.. lbs.
(19) Percentage of ash and refuse in dry coal.......................... per cent
(20) Calorific value of 1 lb. of dry coal by calorimeter test.................. B.t.u.
(21) Cost of coal per ton of .... lbs.. dollars

Hourly Quantities

(22) Dry coal consumed per hour... lbs.
(23) Dry coal per sq. ft. of main fuel bed per hour.......................... lbs.

Indicator Diagrams

(24) Mean effective pressure in lbs. per sq. in...

Speed and Explosions

(25) Revolutions per minute...
(26) Number of explosions per minute...

Power

(27) Indicated horse power developed by engine............................ i.h.p.

Economy Results

(28) Dry coal consumed per i.h.p. per hour.................................... lbs.
(29) Cost of coal per i.h.p. per hour.. dollars
(30) Heat units consumed per i.h.p. per hour (Line 20 × Line 28)............ B.t.u.

Note. For an engine driving an electric generator, the form may be enlarged to include electrical data in the manner given in the code for Complete Steam Power Plants, page 336.

CHAPTER XVII

TESTING OF VENTILATING FANS OR BLOWERS AND AIR COMPRESSORS

Centrifugal Fans are used almost exclusively when large volumes of air are to be handled at a comparatively small pressure. Such a fan consists essentially of a number of plates, either flat or curved, attached to radial arms springing from a central hub through which the driving shaft passes, as in the "spider" type shown in **Fig. 325**, or the blades may be attached to a conical plate as in **Fig. 326**. Fans resembling either of these two designs are known commercially as the "**standard**" type. The "width" of the blades is, in most cases, parallel to the shaft.

Fig. 325. Fig. 326.

"Standard" Types of Ventilating Fans.

The work performed by a centrifugal type of fan is equal to the resistance times the velocity of flow. Since, however, the fan resistances are proportional to the square of the velocity,[1] the work done is proportional to the cube of the velocity.

Disk or Propeller Fans are best illustrated by the so-called "electric" fans so commonly used in offices, shops and dwellings. Fans of this type are usually of a very light construction with the vanes arranged as in a screw propeller for a ship. In many cases fans of this type are not provided with casings, so that it is more difficult to make velocity measurements than with centrifugal fans.

Turbine or "Sirocco" fans have an **impeller** or fan wheel of the "squirrel-cage" type, as illustrated in **Fig. 327**. Fans of this type can be designed to give very high efficiencies. This is due primarily to two characteristic features adopted in these designs. By the use

[1] See Professor Rateau's articles in *Revue de Mécanique* vol. 1, pages 629–837.

of very short blades a very large intake space for the suction is provided which is practically unobstructed, thus giving a very free "suction." The other important feature of this fan is that the air leaves the blades at a higher velocity than that at which the tips of the blades are moving. The importance of this result is shown by a comparison of **Figs. 328** and **329.** The former illustrates the type of blading in a turbine or "Sirocco" fan and shows graphically by a velocity diagram, constructed like a parallelogram of forces, the velocity of the tips of the blades V_b, the velocity of radial flow in the blades V_r and the **absolute** velocity of the discharge V_a, which is the velocity of the air with respect to the stationary casing. It will be observed that in **Fig. 328,** V_a is nearly 50 per cent greater than the velocity of the tips V_b, while in **Fig. 329,** representing the corresponding velocities for a standard type of fan, the

FIG. 327. — Turbine Type (Sirocco) Fan.

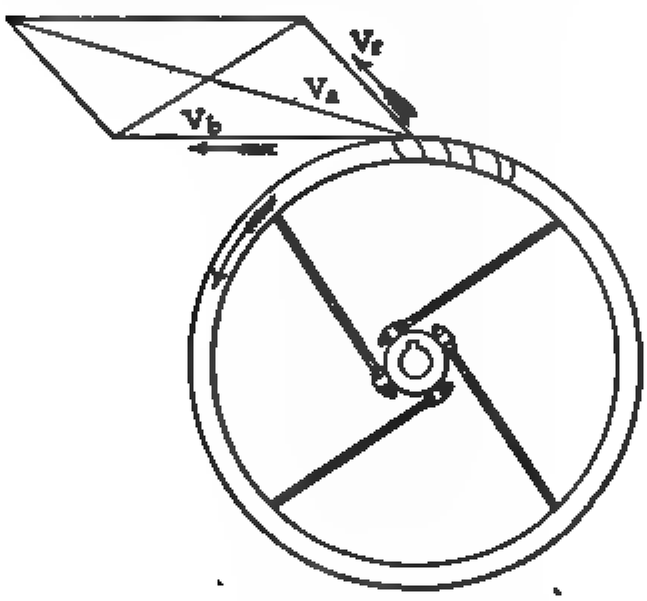


FIG. 328. — Velocity Diagram for a Turbine Fan.

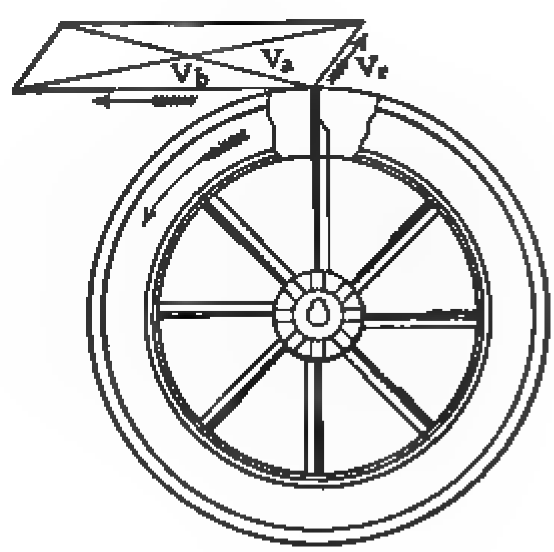


FIG. 329. — Velocity Diagram for a "Standard" Fan.

absolute velocity of the discharge V_a is actually considerably less than the speed of the tips of the blades. Increased velocity is accomplished in a type of fan like **Fig. 328,** not only by the curvature of the tips of the blades but also to some extent by making the blades somewhat concave

with the inner ends (toward the center) practically radial. By this method of designing the outer edges of the blades have a smaller space between them than the inner edges. This has the effect of reducing the area on the discharge side of the blades and consequently the velocity of the air is increased.

Positive Pressure Blowers are used principally for blast furnaces and smelters where a higher pressure is needed than can be efficiently obtained with a centrifugal fan. A section showing the rotors and casing of one of the blowers is shown in **Fig. 330.** It is often called Root's blower. The efficiency of a blower of this type depends on the accuracy of the fitting of the two rotors, **A** and **B**, both with respect to each other and to the casing. It is for this reason that when new the efficiency is high, but after being in service for several years the bearings and the surfaces of the rotors will become worn, so that there is considerable leakage and consequent loss of efficiency.

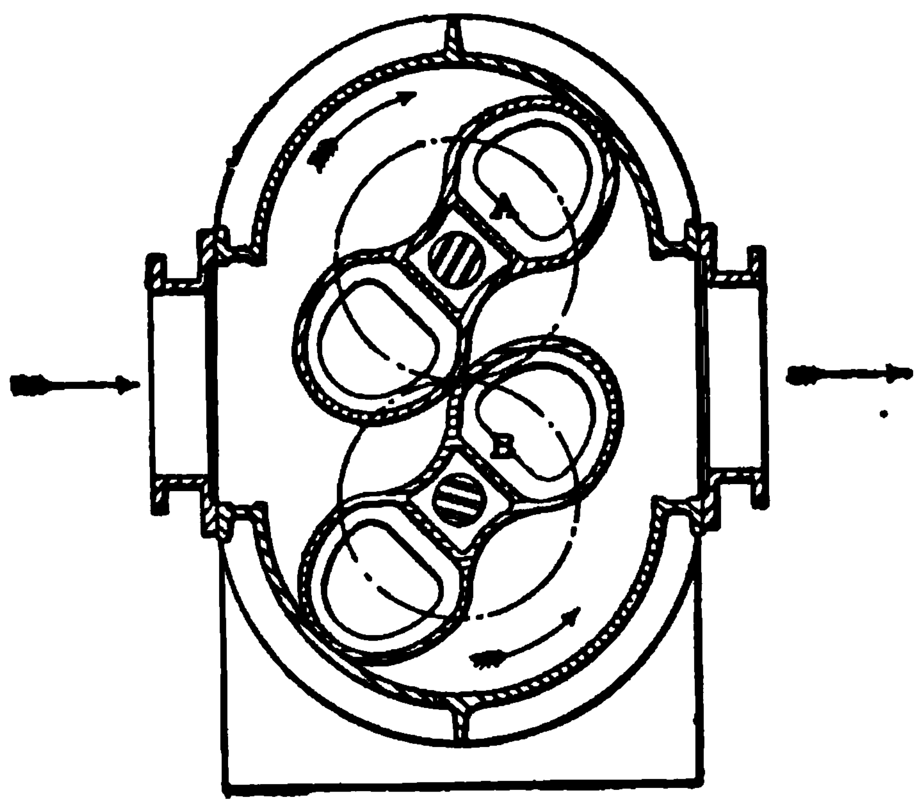


FIG. 330. — Typical Positive Pressure Blower.

Tests of Ventilating Fans or Blowers are made usually by a very simple method; that is, by determining the necessary data for calculating efficiency by measuring the work done by the fan "on the air" in giving velocity, and the power required to drive the fan alone, excluding bearing friction. The fan is preferably operated by an electric motor, of which the efficiency can be readily determined by a Prony brake test. The power required to overcome the bearing friction of the shaft of the fan may well be first determined by measuring the power input[1] (kilowatts) for a series of speeds when the keys fastening the fan to its shaft have been removed and the fan itself has been "blocked" in its casing or, still better, has been removed from the shaft. After attaching the fan again to the shaft the input to the motor and the work done by the fan "on the air" should be determined for various speeds. Then obviously the ratio of the work done by the fan divided by the power required to drive it after correction for the efficiency[2] of the motor and bearing friction is the actual efficiency of the fan. In general terms this may be stated as follows :

[1] For normal operation "friction work" is, for machinery in general, proportional to the speed.

[2] Motor efficiency must be necessarily determined for the conditions of each test; that is, for the same kilowatts and speed as for each test.

f = input to motor to drive motor and shaft of fan in bearings, kilowatts;
i = input to motor to drive motor and fan, in kilowatts;
e = efficiency of motor for motor input of **i** kilowatts and at speed of test;
e′ = efficiency of motor for motor input of **f** kilowatts and at speed of test.

Then if i_n is the net work in **horse power** to drive fan alone,

$$i_n = \frac{ei - e'f}{0.746}. \qquad (108)$$

If the fan to be tested is direct-connected to a steam engine, the test is usually made by measuring the indicated horse power of the engine at the various speeds and also with the fan disconnected for no load. If it is possible to do so the fan should be removed from the shaft for the no-load test to determine bearing friction.

The work done by the fan "on the air" is most readily calculated by the same method used to calculate the efficiency of hydraulic pumps (see page 408), that is, by multiplying the number of pounds of air delivered per unit of time by the head in feet of air corresponding to the discharge pressure. This product is obviously in terms of work in foot-pounds per unit of time, and dividing by 33,000 the corresponding horse power is obtained. Using the following symbols, the same results may be expressed, however, by the product of "pressure times volume" as follows :

v = velocity of air in feet per second;
h = head in feet of air necessary to produce a velocity of **v** feet per second, or,
= water pressure **p** in **inches** observed with a manometer, produced by the **velocity** of the air times the ratio of

$$\frac{wt.\text{ of a cubic foot of water}}{wt.\text{ of a cubic foot of air}^{1}}$$

$$v = \sqrt{2gh} = \sqrt{2g\frac{p \times 62.3}{\text{wt. cu. ft. air for test}}} \qquad (109)$$

and V_m = velocity in feet per minute is (taking $2g = 64.3$)

$$V_m = 1096.4\sqrt{\frac{p}{\text{wt. cu. ft. air for test}}}. \qquad (110)$$

[1] Weight of air taken for calculation must be that corresponding to the TOTAL pressure in the discharge pipe, the temperature and the humidity. For tables of weight of air see pages 180 and 181, also Kent's "Mechanical Engineers' Pocket-Book," 8th edition, pages 583–588, and "Calculating and Testing Ventilating Systems," issued by U. S. Navy Department, Washington.

Now the velocity in feet per minute V_m multiplied by the area of the section at which the velocity was observed, gives cubic feet **C** of air discharged per minute, and if **P** is the total pressure (static + velocity) in pounds per square **foot** then we have for **j** the "**air horse power**" or the work done by the fan "on the air."

$$j = \frac{CP}{33{,}000}, \qquad (111)$$

and efficiency of fan **E**, is

$$E = \frac{j}{i_n} = \frac{CP}{33{,}000\, i_n}. \qquad (112)$$

Velocity measurements are usually made with a **Pitot tube** consisting essentially as shown in **Figs. 224** to **225**, page 179, of two tubes with openings at the end, arranged so that one of them "faces" in the direction of flow and the other extends in a radial direction. The former is subjected to the sum of the velocity and static pressures, while the latter receives only the static pressure.

The following table of relative humidity for determinations with a wet- and a dry-bulb thermometer, **Fig. 331**, or the sling psychrometer, **Fig. 332**, as used by the U. S. Weather Bureau:

TABLE OF RELATIVE HUMIDITY, PER CENT

Dry Thermometer, Deg. F.	Difference between the Dry and Wet Thermometers, Deg. F.																										
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	26	28	30
	Relative Humidity, Saturation being 100. (Barometer = 30 ins.)																										
32	89	79	69	59	49	39	30	20	11	2																	
40	92	83	75	68	60	52	45	37	29	23	15	7	0														
50	93	87	80	74	67	61	55	49	43	38	32	27	21	16	11	5	0										
60	94	89	83	78	73	68	63	58	53	48	43	39	34	30	26	21	17	13	9	5	1						
70	95	90	86	81	77	72	68	64	59	55	51	48	44	40	36	33	29	25	22	19	15	12	9	6			
80	96	91	87	83	79	75	72	68	64	61	57	54	50	47	44	41	38	35	32	29	26	23	20	18	12	7	
90	96	92	89	85	81	78	74	71	68	65	61	58	55	52	49	47	44	41	39	36	34	31	29	26	22	17	13
100	96	93	89	86	83	80	77	73	70	68	65	62	59	56	54	51	49	46	44	41	39	37	35	33	28	24	21
110	97	93	90	87	84	81	78	75	73	70	67	65	62	60	57	55	52	50	48	46	44	42	40	38	34	30	26
120	97	94	91	88	85	82	80	77	75	72	69	67	65	62	60	58	55	53	51	49	47	45	43	41	38	34	31
140	97	95	92	89	87	84	82	79	77	75	72	70	68	66	64	62	60	58	56	54	53	51	49	47	44	41	38

A sling psychrometer is much more accurate than the stationary wet- and dry-bulb type. It should be revolved at about 150 revolutions per minute.

By using the above table the weight of a cubic foot of air for any degree of saturation and temperature can be easily calculated from the tables of the weight of **dry** air and **100 per cent saturated** air as given on page 181.

Anemometers are also frequently used for velocity measurements of air, but they are not generally so reliable as good Pitot tubes. Since,

however, the observations can be taken directly in feet per minute these instruments are used for nearly all work where no great accuracy is expected.

An example showing the method of calculation for **j**, the work done by the fan " on the air," may assist in making the method of calculation clearer.

A series of Pitot-tube measurements taken at ten different places in the cross-section of an air duct shows that the " velocity " pressure was .795 and the total pressure 1.09 inches of water. The observations of barometric pressure and temperatures by wet- and dry-bulb thermometers, together with the " total " pressure given above, served to deter-

FIG. 331. — Wet- and Dry-Bulb Thermometers.

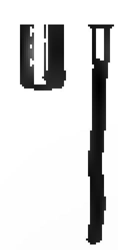


FIG. 332. — Sling Psychrometer. (Wet- and Dry-Bulb Thermometers arranged for Rotation.)

mine the density or the weight of a cubic foot of air at the conditions of the test. Barometric pressure was 39.40 inches of **mercury** and temperatures of wet- and dry-bulb thermometers were respectively 54 and 71 degrees Fahrenheit. According to the tables of the properties of air (see footnote, page 181) this was .07449 pound per cubic foot. Velocity of the air V_m in feet per minute is, therefore,

$$V_m = 1096.4\sqrt{\frac{.795}{.07449}} = \mathbf{3625 \text{ feet per minute.}}$$

The diameter of the pipe was 10 inches, of which the area is 0.545 square foot. Cubic feet air discharged per minute (C) are 3625 × 0.545 or 1978. The total pressure **P** is the " total " pressure[1] in pounds per

[1] In this expression 13.6 is the specific gravity of mercury, .491 is a factor for changing inches of mercury at room temperature to pounds per square inch, and 144 is used to change pounds per square inch to pounds per square foot.

square foot or $\left(\frac{1.09}{13.6}\right).491 \times 144$ or 5.66. Work done by the fan "on the air" is, then,

$$j = \frac{1978 \times 5.66}{33,000} = 0.34 \text{ horse power.}$$

Efficiency tests should be made with the fan operating under the discharge pressure for which it was designed or for which the guarantee was made, as the case may be. The efficiency of the fan may be considerably higher with a lower discharge pressure than when connected up in a ventilating system where the discharge pressure is comparatively high.

Testing Ventilating Systems. When tests are to be made of ventilating systems precautions should be taken in the examination of all ducts and piping to see that they are clear of all lumber, rubbish, etc., and that the dampers are properly set. The tests consist usually in measuring in each system the "static" and the "total" pressures with a Pitot tube with all louvers open. These tests should be made with the fan running at high, low and three or four intermediate speeds. All the results should be checked by plotting a curve with revolutions per minute for abscissas and cubic feet of air delivered per minute for ordinates. This curve should be approximately a straight line passing through the origin if all the louvers have remained open throughout the tests. On the same abscissas curves of pressure for ordinates will also be of advantage in showing the consistency of observations. The location selected for the testing slot to be used for inserting the Pitot tube in the mains should be as near as possible to the fan; preferably no branches, however small, should run off between the fan and the testing slots in the mains. Furthermore the testing slots should not be near turns and bends and particularly no turns or elbows should be immediately ahead of a slot, that is, in the direction toward the fan. These testing slots should be covered when not in use.

In the **U. S. Navy Department the standard conditions** adopted for testing ventilating fans and air-supply mains are a pressure of 5 pounds per **square foot** in the moving air at the discharge outlet from the fan and a velocity of 2000 feet per minute. Air at the standard conditions is to be at 70 degrees Fahrenheit and a relative humidity of 70 per cent. Under these standard conditions a cubic foot of air weighs .07465 pound. The pressure of 5 pounds is equivalent to a **pressure** head of 67 feet of standard density air. A velocity of 2000 feet per minute corresponds to a **velocity** head of 17.27 feet. Total head against which the air is delivered to the supply mains for the standard conditions is then 84.27 feet, making a very satisfactory combination of velocity and pressure head, approaching as it does the maximum possible delivery for this pressure head.

Corrections for Losses of Total Head in Ducts. There is always some loss of total head along a duct or pipe due to friction. As a result there is a smaller delivery than that given for standard conditions. Using the following symbols:

h_f = loss of head in feet due to friction;
f = coefficient of friction = .00008 in piping of good construction;
l = length of duct in feet;
d = diameter of duct in feet;
V_m = velocity of flow through duct in feet per **minute.**

$$h_f = \frac{V_m^2 l}{11{,}250{,}000\, d}. \quad \ldots\ldots \quad (113)$$

If $V_m = 2000$ then $h_f = .3556 \frac{l}{d}$.

Loss of head in a **square duct** is usually assumed to be the same as for a round one; but for a duct of **rectangular section** of which the short side is b and the long side is nb, the formula above becomes, using l and V_m as before,

$$h_f = \frac{1+n}{n} \times \frac{1}{b} \times \frac{V_m^2}{22{,}500{,}000}. \quad \ldots\ldots \quad (114)$$

With the help of these formulas when the size of the main ducts and the discharge in cubic feet per minute at each outlet are known, the head at each outlet as compared with the standard total head of 84.27 feet can be calculated. As a "rough and ready" rule it is often stated that for a loss of one foot of head there is a loss of six-tenths per cent delivery (cubic feet per minute).

Testing Air Compressors. The various types of machines for compressing air are usually operated either by a steam engine or by an electric motor. Power delivered to the compressors by the engine or motor is therefore measured by one of the methods already outlined for ventilating fans. Air compressors, particularly of the reciprocating type, are designed, as a rule, for operation at considerably higher pressures than would be suitable for ventilating fans, and the volume delivered must usually be measured in a comparatively small pipe. Air at high pressure is generally measured by calculating the flow through an orifice in a receiver or one of the other methods described on pages 185 to 188.

Foot-pounds of work "done on the air" per second; and this product divided by the **net power** required to drive the compressor is the "**net**" **mechanical efficiency.** (See also bottom page 373.)

In an air compressor in which the air cylinder is direct-connected to the steam cylinder the net power required to drive the compressor is determined by finding the indicated horse power of the air cylinder and adding the friction in the air cylinder, which in many cases can be as-

sumed to be half the difference between the indicated horse power as measured in the steam and air cylinders. For use on air compressors operating at high pressures special indicators are made. One of the best is made by the Crosby Steam Gage and Valve Co., Boston, Mass., and is illustrated in Fig. 333. It is similar in design to the gas engine indicator illustrated in Fig. 315, page 342, except that the piston in the lower

Fig. 333. — Crosby High-pressure Indicator (Ordnance Type).

cylinder of the indicator is very small, only one-fortieth of a square inch in area. This indicator can therefore be readily used for pressures as high as 10,000 pounds per square inch.

RULES FOR CONDUCTING TESTS OF STEAM-DRIVEN COMPRESSORS, BLOWERS AND FANS[1]

A.S.M.E. CODE OF 1912

If the air end of a compressor is of the reciprocating type, indicator diagrams should be regularly taken from this end as well as from the steam end.

The rules pertaining to dry steam, heat consumption, and indicated horse power of the steam end, are identically the same as those given

[1] In the case of air machinery driven by some other prime mover than a steam engine or turbine, the code may be modified to meet the particular requirements.

on pages 296 and 297 of the Steam Engine Code, and reference may be made to that code for the necessary directions in these particulars.

TESTING OF VENTILATING FANS OR BLOWERS AND AIR COMPRESSORS

(a) *Air Horse Power.* The gross work done at the air end of a reciprocating machine expressed in horse power, is found by multiplying together the net area of the air piston in sq. in., the mean effective air pressure in lb. per sq. in. as determined from indicator diagrams, the length of the stroke in ft., and the number of single strokes per minute; and dividing their product by 33,000.

The net work at the air end of either reciprocating or rotary machines, expressed in ft.-lb. per minute, is found by multiplying the corrected volume of the compressed air in cu. ft. discharged into the main delivery pipe per minute, by the impact or total pressure in lb. per sq. ft. and by the hyperbolic logarithm of the ratio of the total pressure to the atmospheric pressure (all pressures being absolute pressures). The net air horse power is found by dividing the product by 33,000. The corrected volume of the compressed air may be found by multiplying the sectional area of the delivery main in sq. ft. by the mean velocity in ft. per minute as determined by pitot tube or other measurement, and reducing the result to atmospheric temperature by multiplying by the proportion.

$$\frac{460 + t}{460 + T}$$

in which t is the temperature of the air supplied to the machine and T the temperature of the air in the delivery main.

(b) *Capacity.* The capacity is the number of cu. ft. of air discharged through the delivery main per minute, as determined by gasometer, tank or other mode of measurement, reduced to the equivalent free air at the atmospheric temperature and pressure. The correction for pressure is made by multiplying by the proportion $\frac{P_2}{P_1}$ in which P_1 is the atmospheric pressure and P_2 the total pressure in the main (absolute pressures), and the correction for temperature as above.

The capacity may also be expressed in the number of cu. ft. of compressed air discharged per minute at a given pressure above the atmosphere reduced to the atmospheric temperature.

(c) *Miscellaneous.* For methods of calculating results pertaining especially to the performance of the steam-end of a reciprocating air-pumping machine, reference may be made to the Steam Engine Code.

The "efficiency of compression" in a reciprocating machine is determined by first ascertaining the net work at the air end given above under the heading, (a) "Air Horse Power," and then dividing the net work thus found by the gross work given under the same heading.

The "mechanical efficiency" of a reciprocating machine is determined by dividing the gross air horse power at the air end by the indicated horse power at the steam end, or by the horse power delivered by the belt or motor in the case of other means of driving.

DATA AND RESULTS OF TEST OF AIR MACHINERY. CODE OF 1912

(1) Test of.................................located at.........................
to determine........................conducted by...................

(2) Type of machinery..

(3) Rated capacity in cu. ft. of free air per minute. . . . .
(4) Rated capacity in cu. ft. of air discharged per minute at 100 lbs. per sq. in. above atmosphere, reduced to the atmospheric temperature. . . . . cu. ft.
(5) Type of boilers. . . . .
(6) Type of auxiliaries. . . . .
(7) Dimensions of engine or turbine at steam end. . . . .
(8) Dimensions of cylinders or blowers at air end. . . . .
(9) Dimensions of boilers. . . . .
(10) Dimensions of auxiliaries. . . . .
(11) Dimensions of condenser. . . . .
(12) Date. . . . .
(13) Duration. . . . . hrs.

Average Pressures and Temperatures

(14) Steam pressure at boiler by gage. . . . . lbs. per sq. in.
(15) Steam pipe pressure near throttle, by gage. lbs. per sq. in.
(16) Barometric pressure of atmosphere. . . . ins. of mercury = ... lbs. per sq. in.
(17) Pressure in receiver by gage. . . . . lbs. per sq. in.
(18) Vacuum in condenser in ins. of mercury. . . . .
(19) Pressure in delivery main by gage (impact pressure). . . . . lbs. per sq. in.
(20) Total head, expressed in ft. . . . . ft.
(21) Temperature of main supply of feedwater to boilers. . . . . deg. F.
(22) Temperature of additional supplies of feedwater. . . . . deg. F.
(23) Temperature of air in engine room or air supplied to machine. . . . . deg. F.
(24) Temperature by wet-bulb thermometer. . . . . deg. F.
(25) Temperature of air in delivery main. . . . . deg. F.

Total Quantities

(26) Water fed to boilers from main source of supply. . . . . lbs.
(27) Water fed from additional supplies. . . . . lbs.
(28) Total water fed to boilers from all sources. . . . . lbs.
(29) Moisture in steam or superheating near throttle. . . . . per cent or deg. F.
(30) Factor of correction for quality of steam, dry steam being unity. . . . .
(31) Total dry steam consumed for all purposes. . . . . lbs.
(32) Total cu. ft. of compressed air delivered as measured. . . . . cu. ft.
(33) Total cu. ft. of compressed air delivered reduced to atmospheric temperature and pressure. . . . . cu. ft.
(34) Total weight of air delivered. . . . . lbs.

Hourly Quantities

(35) Water fed from main source of supply. . . . . lbs.
(36) Water fed from additional supplies. . . . . lbs.
(37) Total water fed to boilers per hour. . . . . lbs.
(38) Total dry steam consumed per hour. . . . . lbs.
(39) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant. . . . . lbs.
(40) Net dry steam consumed per hour. . . . . lbs.
(41) Dry steam consumed per hour:
(*a*) By main engine or turbine. . . . . lbs.
(*b*) By auxiliaries. . . . . lbs.

(42) Cu. ft. of compressed air delivered per hour as measured................cu. ft.
(43) Cu. ft. of compressed air delivered per hour reduced to atmospheric temperature..cu. ft.
(44) Cu. ft. of compressed air delivered per hour reduced to atmospheric temperature and pressure..cu. ft.
(45) Weight of air delivered per hour..lbs.

HEAT DATA

(46) Heat units per lb. of dry steam based on temperature Line 21............B.t.u.
(47) Heat units per lb. of dry steam based on temperature Line 22............B.t.u.
(48) Heat units consumed per hour based on main supply of feed..............B.t.u.
(49) Heat units consumed per hour based on additional supplies of feed........B.t.u.
(50) Total heat units consumed per hour for all purposes.....................B.t.u.
(51) Loss of heat per hour due to leakage of plant, drips, etc.................B.t.u.
(52) Net heat units consumed per hour..B.t.u.
(53) Heat units consumed per hour:
  (a) By engine or turbine alone...B.t.u.
  (b) By auxiliaries...B.t.u.

INDICATOR DIAGRAMS

(54) Mean effective pressure in steam cylinders.....................lbs. per sq. in.
(55) Mean effective pressure in air cylinders.......................lbs. per sq. in.

SPEED AND STROKE

(56) Revolutions per minute..
(57) Number of single strokes per minute...

POWER

(58) Indicated horse power developed at steam end of reciprocating machine.....i.h.p.
(59) Gross air horse power as indicated in air cylinders of reciprocating machine...air h.p.
(60) Net air horse power as computed from Line 43.............................air h.p.
(61) Friction of reciprocating machine (Line 58 — Line 59).....................fr. h.p.
(62) Percentage of i.h.p. lost in friction of machine.........................per cent

ECONOMY RESULTS, STEAM END OF ENGINE-DRIVEN MACHINES

(63) Heat units consumed per i.h.p. per hour:
  (a) By engine or turbine and auxiliaries.....................................B.t.u.
  (b) By engine or turbine alone...B.t.u.
  (c) By auxiliaries...B.t.u.
(64) Dry steam consumed per i.h.p. per hour[1]:
  (a) By engine or turbine and auxiliaries......................................lbs.
  (b) By engine or turbine alone..lbs.
  (c) By auxiliaries..lbs.

ECONOMY RESULTS, AIR DELIVERED

(65) Heat units consumed per hour per net air h.p. of Line 60:
  (a) By engine or turbine and auxiliaries.....................................B.t.u.
  (b) By engine or turbine alone...B.t.u.
  (c) By auxiliaries...B.t.u.

[1] The i.h.p. on which these economy results are based is that of the main engine given in Line 58.

(66) Dry steam consumed per hour per net h.p. of Line 60:
(*a*) By engine or turbine and auxiliaries . . . . lbs.
(*b*) By engine or turbine alone . . . . lbs.
(*c*) By auxiliaries . . . . lbs.

Efficiency Results

(67) Thermal efficiency ratio for engine alone:
(*a*) Per i.h.p., steam end (2545 ÷ Line 63*b*) . . . . per cent
(*b*) Per net air h.p., air delivery (2545 ÷ Line 65*b*) . . . . per cent

Work Done Per Heat Unit

(68) Ft.-lbs. of net work per B.t.u. consumed by engine or turbine and auxiliaries (1,980,000 ÷ Line 65*a*) . . . . ft.-lbs.

Capacity

(69) Cu. ft. of compressed air delivered per minute as measured . . . . cu. ft.
(70) Cu. ft. of compressed air delivered per minute, reduced to atmospheric temperature . . . . cu. ft.
(71) Cu. ft. of compressed air delivered per minute at 100 lbs. pressure, reduced to atmospheric temperature . . . . cu. ft.
(72) Cu. ft. of compressed air delivered per minute reduced to atmospheric temperature and pressure (free air) . . . . cu. ft.

Miscellaneous Results

Steam-driven Reciprocating Machine

(73) Efficiency of compression (Line 60 ÷ Line 59 × 100) . . . . per cent
(74) Mechanical efficiency of machine (Line 59 ÷ Line 58 × 100) . . . . per cent
(75) Volumetric efficiency (Line 72 ÷ 1st Compr. Displ. × 100) . . . . per cent

Note. In the case of air compressors having more than one stage and in those having intercoolers, additional data should be given covering pressures and temperatures in the different stages, the quantity of water used for cooling and temperatures of the air and water entering and leaving the cooler.

CHAPTER XVIII

TESTING OF REFRIGERATION PLANTS

REFRIGERATING machines present a most interesting example of the conversion of heat energy. In the simplest forms these machines consist of a compressor driven by a steam engine, or other motive power, serving to compress a gas or vapor as the case may be. This gas or vapor is then passed under pressure through a surface condenser, where the cooling water absorbs the heat generated in the work of compression and then passes into an **expanding vessel** into which it discharges at a very low temperature. Now in order to vaporize any liquid, it is necessary to maintain a continual application of heat in order to bring about this physical change. To convert a unit weight of liquid to a unit weight of vapor at the same pressure the heat required is always a constant quantity for the same liquid. Thus, as a familiar example, to convert a pound of water at " atmospheric " pressure and 212 degrees Fahrenheit into steam at the same pressure and temperature requires the application of 970 B.t.u.; and conversely, to condense a pound of steam at this same pressure and temperature, it is necessary to abstract 970 B.t.u. by contact with a cold body. Steam as the working medium in a refrigerating machine would, of course, be impracticable, because the lowest temperature resulting from **actual condensation** in a workable plant would be very much above the freezing point of water; but there are a number of liquids which have a very much lower boiling point than water. Of these ammonia (NH_3), carbon dioxide (CO_2), and sulphurous dioxide (SO_2) are successfully used for purposes of refrigeration. The use of all these depends on the absorption of their latent heat in their conversion from a vapor or gas to the liquid condition. In practice the refrigerating medium most commonly used is anhydrous ammonia, although carbon dioxide is also frequently employed. The latter is preferred usually where ammonia gas might be dangerous or otherwise objectionable.

In the simplest form of refrigerating plant the necessary machinery consists of (1) a **compressor** to raise the gas to the necessary pressure; (2) a **surface condenser** to absorb by means of cooling water the heat generated by the mechanical work of compression; and (3) an **expanding or evaporating vessel** where the liquid is reëvaporated into a gas and, of course, absorbs heat in the operation. A very simple refrigerating machine is shown in **Fig. 334**. It consists of the compressor **C** discharg-

ing gas under pressure[1] through the pipe **P** into the condensing coil **D**, consisting in this simple apparatus of a coil of pipe in a tank through which the cooling water circulates. An expanding valve **V** serves for reducing the pressure and evaporating the liquid coming from **D**. The expanding vessel or **evaporator E** consists of a coil of pipe immersed in a tank containing the liquid to be cooled. Drops of liquid accumulate in the bottom coils of the condenser **D**, to be discharged through the expanding valve **V** into the evaporator **E**. Since the compressor receives its supply of gas from the evaporator, the pressure in the latter must be less than in the condenser. On this account, then, the liquid after expanding will begin to boil and will absorb heat from the surrounding liquid in its transformation into a gas. In such a process the temperature of the cooling liquid may become very low. The refrigerating liquid

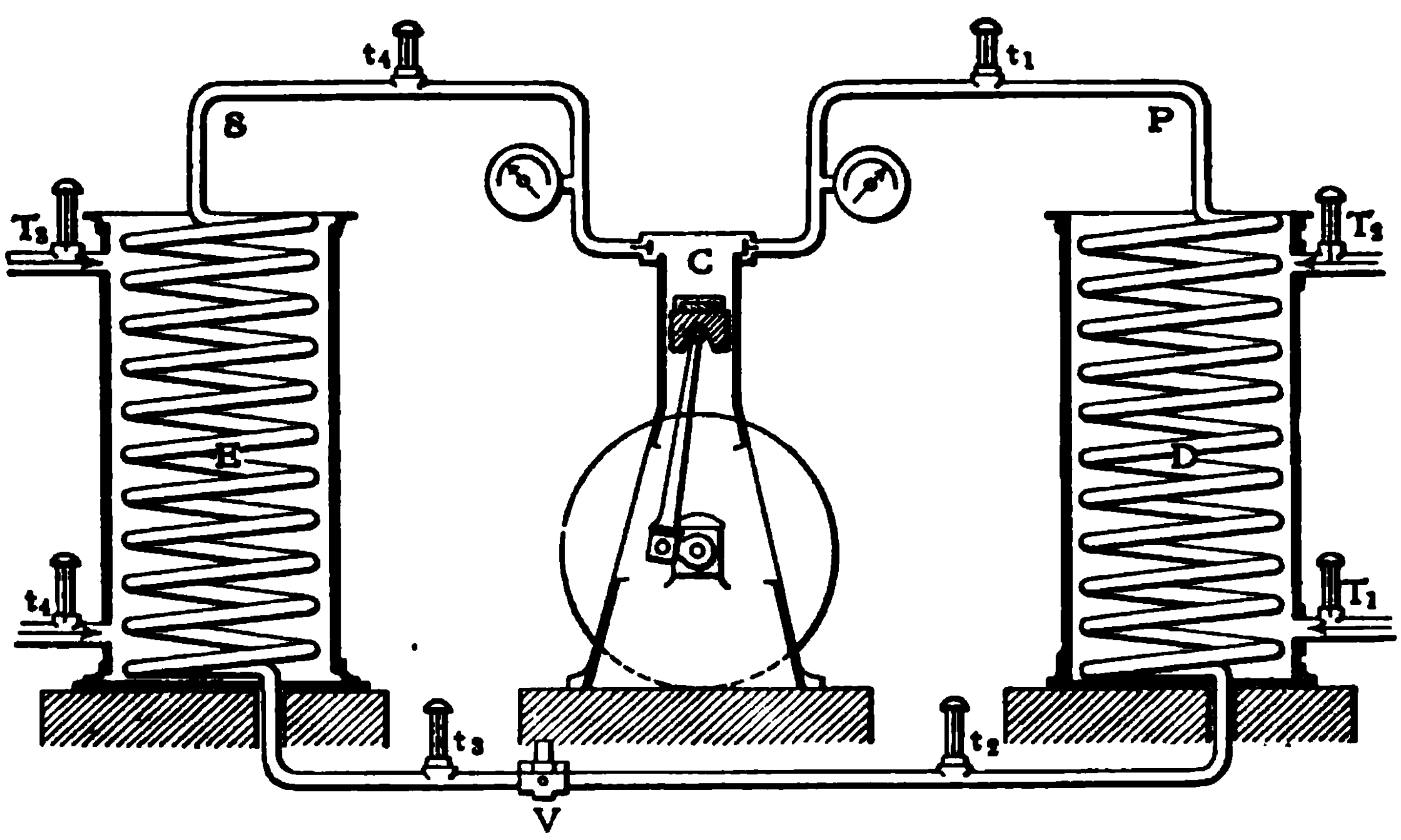


FIG. 334. — Typical Refrigerating Apparatus.

in the evaporator will be entirely gasified or vaporized and returns finally to the compressor **C** in this state through the suction pipe **S**, thus completing the cycle of operations.

After this brief explanation of the principles of the operation of a refrigerating machine we can take up a brief discussion of the thermal processes as regards the interchangeability of heat and work. Using the symbol **r** for the latent heat of vaporization of the refrigerating medium in B.t.u. per pound, **h** for the heat imparted by compression in the same units,[2] **w** for the weight in pounds of the gas or vapor entering the com-

[1] In order to liquefy any gas or vapor, obviously it is necessary to bring the molecules closer together, and this can be accomplished either by increasing the pressure or decreasing the temperature or by both.

[2] The ratio $\frac{r}{h}$ is often called the **coefficient of efficiency** of the refrigerating medium.

pressor in a given time, then, neglecting external losses, **wr** will represent the heat abstracted in the evaporator and **h** + **wr** is the heat given to the cooling liquid in the condenser.

In the practical operation of a refrigerating plant the evaporator is maintained at a very low temperature, and some heat must necessarily be given to it by the refrigerating medium itself, since it enters the evaporator, in comparison, in a moderately warm condition. Now if the difference in temperature between the condenser and the evaporator is **t** degrees Fahrenheit, a pound of refrigerating medium will give to the evaporator **st** B.t.u., if **s** is the specific heat of the refrigerating medium; and further if **w'** is the weight of this medium in pounds passing into the evaporator in a given time then the heat abstracted from the evaporator by the cooling liquid is **wr** − **y** − **w'st**, where **y** is the heat in B.t.u. lost by **radiation.** The term **w'st** is comparatively small in practical machines. If there is no leakage then, of course, **w'** will be the same as **w**.

Anhydrous ammonia is most commonly used as the refrigerating medium. It is preferable to many other fluids because of its comparatively high latent heat[1] and low pressure of vaporization.

Carbon dioxide (CO_2), commercially known as carbonic acid, is a colorless gas without odor when pure, and is furthermore quite innocuous and has practically no injurious effect on animal tissues. It is injurious only when the proportion of it in air becomes so large that there remains an insufficient amount of oxygen. On this account, therefore, it is much safer and suitable as a refrigerating medium than ammonia. This gas can be readily liquefied either by lowering its temperature or by increasing the pressure. At ordinarily low temperatures it can only remain in the liquid state when under considerable pressure. When the pressure is removed, the heat absorbed from surrounding bodies assists in the rapid evaporation of the liquid and these bodies become correspondingly colder by this loss of heat.

Carbon dioxide is used only to a limited extent, but it is found particularly desirable on shipboard because of the compactness of the compressor that it requires and its inoffensive character when a leak occurs.

A typical **commercial refrigerating plant** for making ice and operating with a horizontal ammonia compressor is shown in **Fig. 335.** The same descriptive letters used in **Fig. 334** serve again for marking the important parts.

The efficiency of a refrigerating machine depends upon the difference

[1] The latent heat of vaporization of ammonia is 555 B.t.u. at a temperature of zero degrees Fahrenheit, while that of carbon dioxide is only 123. The corresponding absolute pressures at the same temperature are 30 pounds per square inch for ammonia and 310 for carbon dioxide.

between the extremes of temperature, but unlike heat engines, it has the greatest efficiency when the range of temperatures is small and when the final temperature is high.

When a change of volume of a saturated vapor is made under constant pressure in the presence of an excess of the liquid, the temperature remains constant. In this case the addition or absorption of heat to produce the change of volume causes an increase or decrease in the amount of the liquid mixed with the vapor. Vapors, even when saturated, if no longer in contact with their liquids, having heat added either by compression, by mechanical force or from an external source of heat, will behave practically like permanent gases and will become **superheated.** On this account refrigerating machines using liquefiable

Fig. 335. — Refrigerating Plant with Ammonia Compressor.

gas will give results differing according to the conditions of operation, depending primarily upon the state of the gas; that is, whether it remains constantly saturated or is superheated during a part of the cycle. Some ammonia plants are operated with an excess of liquid present during compression so that superheating is prevented. This is known in practice as the "wet" or "cold" system of compression.

Density of Ammonia Vapor

At temp. deg. C.	−10	−5	0	5	10	15	20
At temp. deg. F. (approx.) ...	14	23	32	41	50	59	68
Density, lb. per cu. ft.	0.6492	.6429	.6364	.6298	.6230	.6160	.6086

Latent Heat of Evaporation of Ammonia

$h_e = 555.5 - .613\,T - 0.000219\,T^2$ (in B.t.u. and degrees F.)

Ledoux found $h_e = 583.33 - .5499\,T - 0.0001173\,T^2$ (in B.t.u. and degrees F.).

For experimental values at different temperatures determined by Professor Denton, see *Transactions American Society Mechanical Engineers*, vol. 12, page 356. For calculated values, see vol. 10, page 646.

SPECIFIC HEAT AND AVAILABLE LATENT HEAT OF HOT AMMONIA

Latent heat at 15.67 lbs. per sq. in. gage press. and 0 degrees F. = 550.5 B.t.u.
Specific heat = 1.096 − 0.0012 T (degrees).

VALUES AT 15.67 LBS. PER SQ. IN. GAGE PRESSURE (LUCKE)

Temperature of Liquid Supply, Deg. F.	Specific Heat of Liquid.	Correction for Cooling, B.T.U.	Available Latent Heat for Saturated Vapor, B.T.U. per lb.
5	1.090	5.45	550.05
10	1.084	10.84	544.66
15	1.078	16.17	539.33
20	1.072	21.44	534.06
25	1.066	26.65	528.85
30	1.060	31.80	523.70
35	1.054	36.89	518.61
40	1.048	41.92	513.68
45	1.042	46.89	508.61
50	1.036	51.80	503.70
55	1.030	56.65	498.85
60	1.024	61.44	494.06
65	1.018	66.17	489.33
70	1.012	70.84	484.66
75	1.006	75.45	480 05
80	1.000	80.00	475.50
85	.994	84.49	471.01
90	.988	88.92	466.58
95	.982	93.29	462.21
100	.976	97.60	457.90

The latent heat of saturated ammonia vapor as given by Lucke must be **corrected** in three ways : (1) For the **temperature of the liquid,** which must be cooled from its initial temperature to the temperature corresponding to the suction or back-pressure; (2) for **wetness of the vapor,** for which the correction is 5.555 B.t.u. for each per cent of moisture; (3) for **superheat of vapor** in case it leaves the expansion coil (evaporator) at a higher temperature than that corresponding to the pressure. This last correction is additive and is approximately the number of degrees of superheat times the specific heat of superheated ammonia gas taken as 0.508.

Leakages of ammonia gas are very objectionable and may be dangerous. One of the most convenient and reliable means for locating a small leak is to burn a little sulphur at the end of a stick of wood about fifteen inches long. Where the sulphur fumes come into contact with the ammonia gas a white vapor is observed.

Units of Refrigeration and Capacity. A practical way to express the performance of a refrigerating plant is found by using as a basis the amount of fuel consumed and the "ice-melting" capacity[1] of the plant. If we use the following symbols:

R = refrigeration of "ice-melting" capacity per pound of fuel, in **pounds**;
w_b = pounds of brine circulated per hour, pounds;
s_b = specific heat of brine;
t_1 = temperature of brine entering expansion coils, deg. F.;
t_2 = temperature of brine leaving expansion coils, deg. F.;
w_f = fuel used per hour, pounds;

$$R = \frac{w_b s_b (t_2 - t_1)}{144\, w_f}, \qquad (115)$$

and the **capacity** C of a machine in tons, **of 2000 pounds,** of **refrigeration** or ice-melting per 24 hours is

$$C = \frac{24\, w_b s_b (t_2 - t_1)}{144 \times 2000}. \qquad (116)$$

Ice-making Capacity is usually defined as half the refrigerating capacity as given by (116).

The above are the "practical" units used in ordinary commercial tests. When, however, facilities are provided for determining the weight of refrigerating medium (ammonia, etc.) a more accurate method of calculation is as follows, using:

c = refrigerating effect per lb. (ammonia, etc.) in B.t.u.;
s = specific heat of liquid (ammonia, etc.);
s_g = specific heat of gas at constant pressure (= 0.508 for ammonia);
t_s = temperature of saturated gas in evaporating coils, deg. F. (from tables);
t_e = temperature of liquid at expansion valve, deg. F.;
t_a = temperature of gas (actual) leaving evaporating coils, deg. F.;
r_s = latent heat of vaporization at temperature t_s;

then
$$c = r_s - s\,(t_e - t_s) + s_g\,(t_a - t_s). \qquad (117)$$

and by this method, capacity of refrigeration, C′ is

$$C' = \frac{W \times c}{288{,}000}, \qquad (118)$$

[1] Ice-melting capacity is a term applied to represent the cold produced in an insulated bath of brine, measured by the latent heat of fusion of ice, which is 144 B.t.u. per pound. More accurately it is the heat required to melt a pound of ice at 32 degrees Fahrenheit to water at the same temperature. The capacity of a machine in pounds or tons of "ice-melting" or of "refrigeration" does not mean that the machine would make that amount of ice; but that the cold produced is equivalent to the melting of the weight of ice to water.

when **W** is weight of refrigerating medium (ammonia, etc.) circulated per 24 hours in pounds.

As calculated by equations (117) and (118) refrigeration units are not comparable for different conditions of operation, and a standard of pressure has come to be quite generally accepted, these being 185 lbs. per sq. in. **gage**[1] pressure at the discharge of the compressor and 15.67 lbs. also by gage and dry saturated gas at the suction. Equivalent refrigerating effect and equivalent tons of refrigeration at these standard conditions are readily calculated from equations (117) and (118), where the last term in (117), representing superheat, becomes zero.

Volumetric Efficiency. The ratio of the actual volume of refrigerating medium, discharged from the compressor to that calculated from the piston displacement is called the volumetric efficiency. The following formula deduced from Voorhees[2] gives in most practical cases the volumetric efficiency $\mathbf{E}_v$ of an ammonia compressor with a remarkable degree of accuracy:

$$\mathbf{E}_v = 1 - \frac{(\mathbf{t}_1 - \mathbf{t}_0)}{1330}, \qquad (118)$$

where $\mathbf{t}_1$ is the **theoretical** temperature of the gas after compression, $\mathbf{t}_0$ is the temperature of the gas delivered to the compressor. Here $\mathbf{t}_0$ can be calculated from the general equation for adiabatic compression where

$$\mathbf{t}_1 + 460 = (\mathbf{t}_0 + 460)\left(\frac{\mathbf{p}_1}{\mathbf{p}_0}\right)^{0.24} \qquad (119)$$

Here $\mathbf{p}_1$ and $\mathbf{p}_0$ are the absolute pressures of the gas corresponding respectively to the temperatures $\mathbf{t}_1$ and $\mathbf{t}_0$. The actual temperature of the gas discharged from the compressor will be usually considerably, sometimes from 50 to 60 degrees Fahrenheit less than the theoretical.

Lucke[3] has deduced the following formula for the **indicated horse power of compressor (i.h.p.) required per ton of refrigerating capacity,** expressed in the following symbols:

$\mathbf{p}$ = the mean effective pressure in compressor in lbs. per square inch;
$\mathbf{l}$ = the length of the stroke in feet;
$\mathbf{a}$ = the area of the piston in square inches;
$\mathbf{n}$ = the number of compressions per minute;
$\mathbf{E}_v$ = the volumetric efficiency, as defined above;
$\mathbf{w}_c$ = the weight of a cubic foot of ammonia vapor at the back pressure as it exists in the cylinder when compression begins; $\mathbf{v}_c$ is the latent heat of vaporization available for refrigeration at suction pressure (see table

[1] Since gage pressures are used it is obvious our methods of calculation for refrigerating machinery are not on a sound scientific basis.

[2] "Ice and Refrigeration" (1902).

[3] *Proceedings American Society of Refrigerating Engineers* (1908).

page 381); 288,000 = the B.t.u. equivalent to one ton of refrigeration per twenty-four hours, that is, 2000 × 144. Then,

$$\text{i.h.p.} = \frac{\dfrac{\text{plan}}{33{,}000}}{\dfrac{\text{laE}_v\text{nw}_c \times \text{v}_c \times 60 \times 24}{144 \times 288{,}000}}. \quad . \quad . \quad . \quad . \quad (120)$$

$$= \frac{0.87}{\text{w}_c\text{v}_c} \times \frac{\text{p}}{\text{E}_v}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (121)$$

Theoretical Efficiency of Refrigerating Machines. The maximum theoretical efficiency $\mathbf{E_m}$ of a refrigerating machine is expressed by the ratio,

$$\mathbf{E}_m = \frac{\mathbf{T}_0}{\mathbf{T}_1 - \mathbf{T}_0}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (122)$$

where $\mathbf{T}_1$ is the highest and $\mathbf{T}_0$ is the lowest absolute temperature of the refrigerating medium.

Heat Balance. The heat balance for the cycle of operation may be calculated as follows: The liquid enters the brine coils at a temperature of $\mathbf{t}_e$; **first**: this temperature must be lowered to $\mathbf{t}_s$ the temperature of saturation. Under ideal conditions (without losses) the heat added to the brine by this lowering of the temperature of the liquid is $\mathbf{s}(\mathbf{t}_e - \mathbf{t}_s)$ per pound of liquid. **Second,** the liquid is vaporized and takes the latent heat of the ammonia $\mathbf{r}_s$ per pound of liquid from the brine. **Third,** if the temperature of the gas leaving the coils is superheated then this heat received by the gas from the brine in addition to the above will be $\mathbf{s}_g(\mathbf{t}_a - \mathbf{t}_s)$ B.t.u. per pound of ammonia. Then the **total heat $\mathbf{H}_1$ (B.t.u.) received by the ammonia per pound** in passing through the brine coils will be (under ideal conditions — no losses), using symbols as on page 382,

$$\mathbf{H}_1 = \mathbf{r}_s - \mathbf{s}(\mathbf{t}_e - \mathbf{t}_s) + \mathbf{s}_g(\mathbf{t}_a - \mathbf{t}_s). \quad \text{[Same as (117).]}$$

This will also represent the amount of **heat taken from the brine tank** per pound of ammonia circulated.

Heat received in passing from the brine tank to the compressor in B.t.u. per pound of ammonia gas circulated, where $\mathbf{t}_1$ is the temperature of the gas at the compressor before entering, deg. F.,

$$\mathbf{H}_2 = \mathbf{s}_g(\mathbf{t}_1 - \mathbf{t}_a).$$

Heat Received in Passing through Suction Valves. The gas in passing **through the suction valves** of the compressor is superheated due to the friction and throttling when coming in contact with the hot metal in the cylinder. The temperature $\mathbf{t}_2$ at the beginning of compression can be determined approximately as follows:

$$\mathbf{t}_2 = \mathbf{t}_1 + \frac{12}{\mathbf{d}} \times 55.5 = \mathbf{t}_1 + \frac{666}{\mathbf{d}},$$

when **d** is the diameter of the compressor in inches. Then heat added in the valves $\mathbf{H}_3 = \mathbf{s}_g(\mathbf{t}_2 - \mathbf{t}_1)$ per pound of ammonia circulated.

Heat due to Compression. The work done as shown by the indicator card will represent the heat added to the gas by the compressor. The work done on the gas is represented by the total work shown by the indicator cards to be taken from all the cylinders $\mathbf{W}_k$, which is in B.t.u. per pound of ammonia.

Total Heat Added. Total heat added to the ammonia per pound of liquid flowing through the cycle will be

$$\mathbf{H} = \mathbf{H}_1 + \mathbf{H}_2 + \mathbf{H}_3 + \mathbf{W}_k.$$

Heat by Jacket Water. The water flowing through the jackets of the compressor cylinder removes some heat from the ammonia gas, which is, $\mathbf{H}_4 = \mathbf{w}_1(\mathbf{t}_5 - \mathbf{t}_4)$ where $\mathbf{w}_1$ is weight of jacket water circulated in same time as one pound of ammonia, also $\mathbf{t}_5$ and $\mathbf{t}_4$ are respectively temperatures of water leaving and entering, deg. F.

Heat Absorbed by Ammonia Condenser: $\mathbf{H}_5 = \mathbf{r}_c + \mathbf{s}_g(\mathbf{t}_3 - \mathbf{t}_c)$ in B.t.u. per pound of liquid. Where $\mathbf{t}_c$ = temperature of condensed liquid in ammonia receiver, deg. F., $\mathbf{r}_c$ = the corresponding latent heat, and $\mathbf{t}_3$ = temperature of gas as discharged from the compressor.

Heat absorbed by condensing water $\mathbf{w}_2(\mathbf{t}_7 - \mathbf{t}_6)$ should equal the above when $\mathbf{w}_2$ is weight of condensing water used while one pound of ammonia is circulated, also $\mathbf{t}_7$ and $\mathbf{t}_6$ are the temperatures of water leaving and entering, deg. F.

Heat Balance. Heat added = heat absorbed + radiation (**R**).

$$\mathbf{H}_1 + \mathbf{H}_2 + \mathbf{H}_3 + \mathbf{H}_w = \mathbf{H}_4 + \mathbf{H}_5 + \mathbf{R}. \qquad (123)$$

Mechanical Efficiency. The mechanical efficiency of the machine will be the ratio between the work done in the compressor cylinder to that done in the steam cylinder.

$$\text{Mech. Eff.} = \frac{\text{i.h.p. compressor}}{\text{i.h.p. engine}}. \qquad (124)$$

Ammonia Absorption Refrigerating Machines. Another class of refrigerating apparatus, operating by what is known as the absorption system, has been installed in some places. It consists of a generator containing a concentrated solution of ammonia in water. This generator is heated usually by means of a coil of pipes taking live steam from a boiler, although frequently the exhaust steam from engines is utilized to advantage. In this system a weak ammonia vapor passes first into an "analyzer" where some of the water is separated from the ammonia vapor and then into a "rectifier," where the concentrated vapor is cooled, precipitating still more water, and then discharges into the condenser coils. The lower coils of the condenser are connected to the upper part of the "cooler" or brine tank. An absorption chamber is provided which is filled with a weak solution of ammonia, and this

chamber is also connected with the cooling tank. The absorption chamber communicates with generator by two tubes, one going to the bottom of the generator from the top of the chamber, and the other from the bottom of the chamber to the top of the generator. In the latter pipe line, a pump is located to force the liquid from the absorption chamber, where the pressure is about atmospheric, to the generator, where the pressure is from 100 to 200 pounds per square inch. In the operation of this apparatus the ammonia and water in the generator are first heated by the coil of steam pipes, and as the ammonia is freed from the solution the pressure rises. When this pressure attains that of saturated vapor at the temperature of the condenser it becomes liquefied, condensing also a small amount of steam. A suitable expansion valve regulates the flow of the liquefied gas into the refrigerating coils in the cooler. As it escapes into these coils it expands and is again vaporized, absorbing heat from the liquid or gas required to be cooled. Just as rapidly as vaporization goes on the gas is absorbed by the weak solution in the absorbing chamber. The heat in the generator has the effect of separating a strong from a weak solution, the greater concentration being in the upper part. The weaker portion of the solution is conveyed by the pipe entering the top of the absorption chamber. The satisfactory operation of this apparatus depends upon careful adjustments and regulation of the flow of gas and liquid, controlling in this way the temperature in the cooler.

Testing of Refrigerating Plants. The primary object of a test of a refrigerating apparatus is to compare the refrigerating effect with the heat equivalent of the mechanical work and of the cooling of the water or brine. The making of ice is not satisfactory for accurate results in a test. The range of temperature should not be greater than necessary to secure accuracy in the thermometer readings. The brine should be measured or weighed in suitable tanks as for the condensed steam in engine tests.

One of the most important precautions to be observed is to determine accurately the specific heat of the brine for the temperature range of the test. Small differences in its concentration and composition may produce a considerable variation in results. When a compressor and steam engine are coupled directly together on the same shaft a direct measurement of the power required for the compresser is not obtainable. By measuring the horse power of the engine running without doing any work in the compressor — that is, operating it "empty" — and by comparing the differences in power between the steam engine and compressor for wide variations of condenser pressure, the effective horse power required to drive the refrigerating machine can be determined with some degree of accuracy.

The following data sheet is used in parts by Denton[1]:

1. Average high ammonia pressure above atmosphere..........................
2. Average back ammonia pressure above atmosphere..........................
3. Average temperature brine inlet..........................
4. Average temperature brine outlet..........................
5. Average range of temperature..........................
6. Lbs. of brine circulated per minute..........................
6a. Specific heat of brine..........................
7. Average temperature condensing water at inlet..........................
8. Average temperature condensing water at outlet..........................
9. Average range of temperature..........................
10. Lbs. water circulated per minute through condenser..........................
11. Lbs. water per minute through jacket..........................
12. Range of temperature in jackets..........................
13. Lbs. ammonia circulated per minute..........................
14. Probable temperature of liquid ammonia entrance to brine-tank..........................
15. Temperature ammonia corresponding to average back pressure..........................
16. Average temperature of gas leaving brine tank..........................
17. Temperature of gas entering compressor..........................
18. Average temperature of gas leaving compressor..........................
19. Average temperature of gas entering condenser..........................
20. Temperature due to condensing pressure..........................
21. Heat given ammonia:
    By brine per B.t.u. per minute..........................
    By compressor, B.t.u. per minute..........................
    By atmosphere, B.t.u. per minute..........................
22. Total heat received by ammonia, B.t.u. per minute..........................
23. Heat taken from ammonia:
    By condenser, B.t.u. per minute..........................
    By jackets, B.t.u. per minute..........................
    By atmosphere, B.t.u. per minute..........................
24. Total heat rejected by ammonia, B.t.u. per minute..........................
25. Difference of heat received and rejected, B.t.u. per minute..........................
26. Per cent of work of compression removed by jackets..........................
27. Average revolutions per minute..........................
28. Mean effective pressure steam cylinder, lbs. per square in..........................
29. Mean effective pressure ammonia cylinder, lbs. per square in..........................
30. Average H.P. steam cylinder..........................
31. Average H.P. ammonia cylinder..........................
32. Friction in per cent of steam H.P..........................
33. Total cooling water, gallons per minute, per ton ice-melting capacity per 24 hours.
34. Tons ice-melting capacity per 24 hours..........................
35. Lbs. ice-refrigeration effect per lb. coal at 3 lbs. per H.P. hour..........................
36. Cost coal per ton of ice-refrigerating effect at $4 per ton..........................
37. Cost water per ton of ice-refrigerating effect at $1 per 1000 cu. ft. per 24 hours...
38. Total cost of 1 ton ice-refrigeration effect..........................
39. Refrigeration effect per I.H.P. in compress. cyl., B.t.u. per minute..........................
40. Refrigeration effect per I.H.P. in steam, cyl., B.t.u. per minute..........................
41. Refrigeration effect per pound of steam, B.t.u. per minute..........................

[1] *Transactions American Society of Mechanical Engineers*, Vol. 12, page 356.

Another form for data and results, including a heat balance is as follows:

RESULTS OF REFRIGERATING PLANT TESTS

General

1. Date of test.........................located at..........................
2. Object of test...
3. Duration of test..
4. Type of machine...
5. *Dimensions:*
6. Diam. of ammonia cyl.............diam. steam cyl........................
7. Stroke of compressor.......stroke of steam engine...........................
8. Diam. of piston rod comp. cyl. .... diam. piston rod steam cyl................
9. *Average Temperatures of Ammonia:* Leaving machine.... entering machine.......
10. Leaving condenser .... entering expansion coils...............................
11. Leaving expansion coils..
12. *Average temperatures of water:*
13. Entering ammonia condenser .... leaving ammonia cond......................
14. Entering compressor jackets .... leaving comp. jackets......................
15. *General temperatures:* All temperature in degrees...........................
16. Room .... outside air .... brine...
17. *Pressures:* (Lbs. per sq. inch absolute.)
18. Steam .... discharge, ammonia .... suction ammonia.........................
19. Atmospheric...
20. *Weights:* (pounds.)
21. Ammonia used .... jacket water .... ammonia condenser..................
22. Ammonia per hour .... jacket water per hour...............................
23. Ammonia condensed per hr. .... ammonia per 24 hrs........................
24. Jacket water per 24 hrs. .... ammonia cond. per 24 hrs......................
25. Scale of spring, ammonia cylinder.............................lbs. per sq. in.
26. Scale of spring, engine cylinder...r.p.m.
27. Average i.h.p. per hour (steam cylinder)..
28. Average i.h.p. per hour (ammonia comp.).......................................
29. Condition of gas leaving machine................................deg. F. sup.
30. Condition of gas at beginning of compression......................deg. F. sup.
31. Condition of gas entering machine..............................deg. F. sup.
32. Refrigerating effect, actual, per lb. ammonia...........................B.t.u.
33. Tons of refrigeration, actual...
34. Ice-making capacity, actual (line 33 ÷ 2)......................................tons
35. Mechanical efficiency...per cent
36. Equivalent refrigerating effect based on standard conditions (see page 383) B.t.u. per lb.
37. Equivalent tons of refrigeration based on standard conditions (see page 382).....
38. Ice-making capacity based on standard conditions.........................tons
39. Per cent rating obtained..
40. Volumetric efficiency (see page 383).......................................per cent
41. Heat balance (see pages 384)..

CHAPTER XIX

TESTING OF HOT-AIR ENGINES

Hot-air Engines of the conventional type are reciprocating " piston " engines which are operated by the alternate expansion and contraction of a charge of air. This alternate expansion and contraction is produced by heating and cooling. Engines of this kind now in use are found most often in country places, where they are used for pumping water. Usually coal is burned for fuel; but sometimes gas is used, particularly in the natural-gas districts.

Rider Hot-air Engine. The most successful engine of this kind is made by the Rider-Ericsson Engine Company of New York. This engine, illustrated in Fig. 336, consists of a compression cylinder **C** and a power cylinder **P**, each provided with a separate piston. These two cylinders are connected together by a rectangular passage **R** containing a large number of thin metallic plates and forming what is called in engines of this type the **regenerator.** This regenerator has for its function the alternate abstracting and returning to the air of a quantity of heat. Air leaking out is replaced by a fresh supply admitted through the check valve **V**, which opens inward. The compression cylinder **C** is provided with a water-jacket.

C–Compression Cylinder
P–Power Cylinder
E–Cooler
H–Heater
R–Regenerator
II–Cranks set at about 100°
JJ–Connecting Rods
L–Check Valve
M–Pump Primer
T–Water Jacket, to protect packing from heat
U–Pump

FIG. 336. — Hot-air Engine.

The cycle of operations in this engine consists of a compression stroke when the piston in the compression cylinder **C** compresses the cold

air from which the heat has been abstracted by its passage through the regenerator **R**, and then by the simultaneous advancing upward movement of the piston in the power cylinder **P** the air passes again through the regenerator and also through the **heater H** without appreciable change of volume. As a result the addition of heat increases the pressure of the air and when it enters the power cylinder **P** it pushes the piston upward to the end of its stroke. This upward movement of the power piston in the last half of its stroke carries with it the piston in the compression cylinder **C**, which is on the same shaft but set at an angle of 90 degrees, so that the two pistons do not reach the ends of their strokes together. Now as the charge of air cools the pressure falls, so that the piston in the power cylinder falls and in the last half of this stroke carries downward with it the piston in the compression cylinder and again starts compressing the charge of air. As the heated air passed through the regenerator plates on its way to the compression cylinder the greater portion of the heat it contained was left in them to be abstracted on the return movement to be used again for increasing the temperature of the charge.

In **Fig. 336** a water pump **U** is shown at the left-hand side of the engine. Besides being used for supplying a water system it pumps the cooling water needed for the water-jacket **T** and the cooler **E**.

Tests of Hot-air Engines do not differ in the important details from tests of steam and gas engines. The indicated horse power is obtained by attaching engine indicators to both the power and the working cylinders and the net indicated horse power is the difference between that for the power and that for the compression cylinders. A Prony brake or similar device attached to the main shaft for absorbing the power can be used to determine the useful or brake horse power and the ratio of the brake to the indicated horse power is the mechanical efficiency.

For testing such engine to determine the efficiencies and economy it is preferable to use gas or oil for fuel instead of coal, because of the obvious advantages in the determination of fuel consumption.

Thermodynamic efficiency is the ratio of the range of temperature to the initial absolute temperature of the air in the power cylinder. Temperatures not determinable by direct measurement may be calculated from the pressures and the specific volumes by the general formula for perfect gases,

$$T = pv/R, \qquad (122)$$

where **R** for air is 53.21.

CHAPTER XX

TESTS OF HOISTS, BELTS AND FRICTION WHEELS

Efficiency of Hoists. An efficiency test of a hoist is made by determining the ratio between the work done in lifting the load to that applied to the hand chain. Stated briefly, this determination is made by raising slowly a known weight, observing at the same time by means of a spring balance fastened to the hand chain, the pull or force required to keep the load moving after it has been started. The method is, of course, the same for determining the efficiency of a rope hoist, except that some special provision must be made for attaching the hook of the spring balance to the rope. Allowance must also be made, of course, for the number of times the power is multiplied, that is, the relative velocity value.

Differential Hoists. Differential hoists (Fig. 337) are a little more complicated than the ordinary chain or rope hoist. In this apparatus, as shown in the standard books on the theory of mechanism, the velocity ratio is expressed according to the dimensions in the figure by

$$\frac{2\,R_1}{R_1 - R_2}. \qquad (123)$$

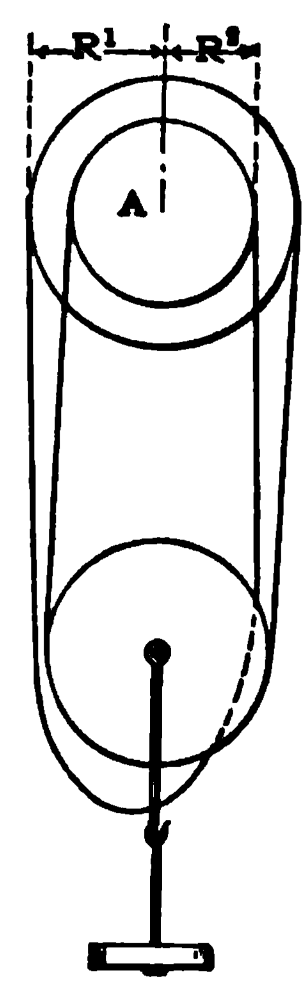


Fig. 337. — Differential Hoist.

It is difficult, however, to measure accurately the radius of these wheels on account of the irregular surface made for gripping the links. Now since the circumferences of these wheels are proportional to the radii, the velocity ratio may be determined by counting the number of link-pockets in each of the wheels, and its value will be given by the ratio of twice the number of link-pockets in the larger wheel divided by the difference between the number of link-pockets in the larger and the smaller wheels. In some other types of hoists where this method is not applicable and the diameters cannot be readily measured, the velocity ratio can be determined by tying a piece of string on a link of the "load" chain or rope, as the case may be, opposite some fixed part of the hoist and mark in the same way a point on the chain or rope to which the pull is applied. Now when the "load" chain has been moved a measured distance, the corresponding move-

ment of the point of application can be measured. Several observations should be made to eliminate probable errors in the measurements. Velocity ratio is usually expressed by making the second member of the ratio unity, thus 13 : 1, 4 : 1, etc.

The force required to move the hand chain by the spring balance multiplied by the velocity ratio is the work "put into" the hoist, while the weight lifted is proportional to the work done. Efficiency is then the work done divided by the work "put in," or in the terms above is the weight lifted divided by the product of the pull on the hand chain times the velocity ratio.[1] On account of the great friction at starting the reading of the spring balance should be made when constant after starting by hand. Determinations should also be made when the load is being lowered, but these results should not be averaged with those for raising the load because a hoist is generally used only for raising loads.

Determination of Tension in Belts and Rope Drives. Tests are often required to determine the power transmitted by belts and ropes for specified conditions of load, speed, tension, and the coefficient of friction between these and the pulleys on which they run. Suitable apparatus for such tests consists of a device with which the belts or ropes can be operated with different tensions. Usually the load is not increased beyond the limit producing 3 per cent slip.[2] The initial tension in the belt or rope should be measured when at rest.

Testing of Belts. For determining the qualities of belts as regards power transmission and the coefficient of friction belt tests are desirable. Sometimes belts or pulleys of different materials are to be compared, while again tests may be required to determine the effect of belt dressings. The apparatus to be described is simple in construction and inexpensive. It consists essentially of two pulleys on the ends of a universal coupling. One of these pulleys is driven by a variable speed motor to which it is belted. The other is set up on the frame shown in **Fig. 338** suspended on a knife-edge bearing at O so that the end at S is free to move in a vertical plane and any horizontal tension in a belt placed on it produces a proportional pressure on a scales at S. This pulley shown in the figure is the driver for another pulley attached to shafting about

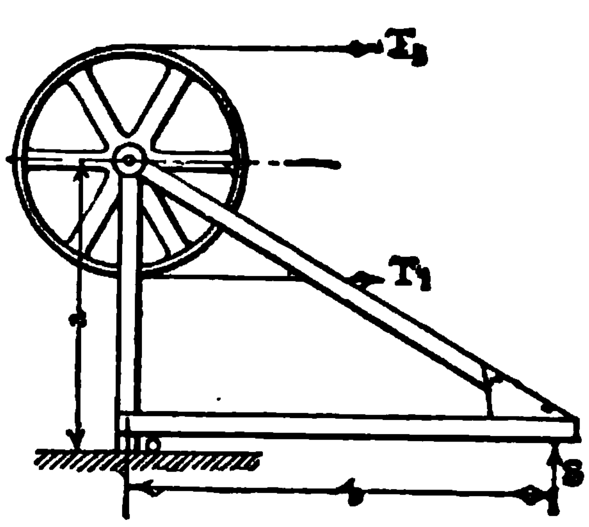


FIG. 338. — Belt and Rope Testing Apparatus.

[1] If the spring balance is used in the inverted position, its weight must be added to the pull. Also the weight of the scale pans used for supporting the weights must be added.

[2] Slip in belts or ropes is ratio of the difference between the revolutions of the driver and the follower divided by the revolutions of the driver, each taken, of course, for the same time unit.

25 feet away in fixed bearings and is attached to a Prony brake. This last shaft is adjustable in its bearings so that the tension in the belt can be varied.

Let T_1 = tension in tight side of belt, lbs.;
T_2 = tension in slack side of belt, lbs.;
W = pressure on scales at S, lbs;

then
$$(T_1 + T_2)\,a = Wb \quad \text{or} \quad T_1 + T_2 = \frac{Wb}{a}. \qquad (124)$$

When power is being transmitted by the belt and absorbed by the Prony brake we have, since the power transmitted is proportional to the difference of tension, by moments $wR + T_2r = T_1r$, and then

$$T_1 - T_2 = \frac{wR}{r}, \qquad (125)$$

where

w = net weight on brake, lbs.;
R = length of brake arm, ft.;
r = radius of driven pulley, ft.

Combining (124) and (125)

$$T_1 = \frac{\frac{Wb}{a} + \frac{wR}{r}}{2}. \qquad (126)$$

$$T_2 = \frac{\frac{Wb}{a} - \frac{wR}{r}}{2}. \qquad (127)$$

These last equations give the **total** tensions T_1 and T_2 which are usually reduced to pounds per inch of width of the belt.

The **coefficient of friction** (f) between the belt and the pulley is given in books on Mechanics as

$$f = \frac{\log \frac{T_1}{T_2}}{0.434\, c}, \qquad (128)$$

where c is the arc of contact of the belt on the pulley in inches divided by the radius of the pulley in inches.

Belts will be stretched more on the tight than on the slack side and this unequal stretching causes a difference in velocity, or per unit of time, a difference between the length of belt on the driving and driven pulleys. This difference is called the **creek** or **slip** of the belt. If r' is the radius and N' the r.p.m. of the driving pulley, while r is the radius and N is the r.p.m. of the driven pulley then the difference in velocity is $2\pi r'N' - 2\pi rN$. This is because the length of belting coming on a

pulley in a unit of time is equal to its peripheral speed. Slip or creek is expressed usually as a percentage of the speed of the driver so that

$$\text{Percentage slip or creek} = \frac{r'N' - rN}{r'N'}. \quad . \quad . \quad . \quad .(129)$$

Efficiency of Transmission = Delivered Horse Power ÷ Horse Power Input.

Tests of Friction Wheels. The apparatus frequently used for determining the **coefficient of friction** between friction wheels consists of a pair of pulleys, one of them at least usually made of some soft metal like aluminum or a fibrous material like straw fiber, leather fiber or paper. This driver runs on a follower generally of some other material. Power delivered to the "follower" shaft is absorbed by the Prony brake. A bell-crank lever to which weights can be attached is used to hold or press the two pulleys together.

The coefficient of friction as determined by such an apparatus is the ratio of the tangential pull to the total normal pressure. If the coefficient of friction is represented by **f**, and other symbols are used as follows:

r_1 = the effective brake arm in inches;
r_2 = the radius of driven pulley;
w = net weight on the brake in pounds;
p = normal pressure in pounds per inch of width;
z = width of the narrower pulley in inches;

then,

$$f = \frac{\frac{wr_1}{r_2}}{pz} = \frac{wr_1}{pr_2z}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (130)$$

CHAPTER XXI

TESTING OF LUBRICANTS

In the space which can be taken for this chapter on the testing of oils, only a few of the simpler and more important tests can be given. To make a complete report on the composition of a sample of oil it is generally advisable to submit the suspected sample to a competent professional chemist, preferably with a large and varied experience in the analysis of lubricants. Characteristic tests, however, will be given here which should be sufficiently serviceable to engineers to determine whether an oil is sufficiently pure for the use intended and whether, if it is to be used as a lubricant, it has the properties essential for reducing friction to a minimum, or if used as a fuel, it is necessary to ascertain the degree to which it is susceptible to vaporization, and in the case of very light oils, the temperature to which it may be exposed without danger of becoming inflammable. The **heating or calorific value of fuel oils** is also a very important test, but this has been explained in Chapter VIII. It remains therefore only to take up for fuel oils the tests for **specific gravity** and the **flash and burning points.**

Specific gravity of a liquid is determined most accurately in most cases where a sensitive chemist's balance is available, by the use of a specific gravity bottle. A conventional type is shown in **Fig. 340.** Bottles for this purpose are made with special care of thin glass and the weight of distilled water which they will contain is determined accurately and etched on the outside surface of the bottle. The bottle is provided with a small " ground " glass stopper having a capillary tube or hole drilled through it, so that when the bottle is filled to the top of the capillary tube it will always hold the same volume of liquid.

In the operation of determining the specific gravity the bottle is filled with the liquid to be tested being careful to avoid the formation of air bubbles. The stopper is then inserted and some of the liquid will run out through the capillary tube. This excess should be wiped off so that the bottle will be clean and dry. It can then be weighed. After once determining the weight of the bottle filled with distilled water at 60 degrees Fahrenheit the bottle can be used without again weighing it with water.[1] The weight of the empty bottle should be ascertained from

[1] This suggestion is made because it is not always easy for engineers to obtain clean distilled water. Condensed steam from a surface condenser is usually, however, sufficiently free from impurities, and is, of course, distilled water.

time to time to determine, more than for any other reason, whether it is clean. If it is found to weigh more than when new, obviously it needs cleaning. The weight of the liquid in the bottle when it is full divided by the weight of the corresponding amount of distilled water is the specific gravity of the liquid being investigated. The liquid tested should be at 60 degrees Fahrenheit when it is weighed in the bottle, as this is the standard temperature for the specific gravities of all oils.

For the determination of the specific gravity of very thick oils and greases, a type of bottle or tube known as Hubbard's (**Fig. 341**) is often used. It consists of a metallic tube with a ground-in stopper, having a slightly larger bore than the capillary tube in the glass stopper of the ordinary specific gravity bottle. In commercial practice the specific

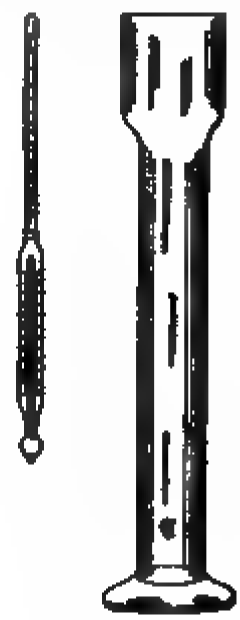


FIG. 340. — Specific Gravity Bottle for Thin Oils.

FIG. 341. — Specific Gravity Bottle for Thick Oils.

FIG. 342. — Hydrometer and Hydrometer Jar.

gravity of liquids is usually determined by means of an instrument called a **hydrometer, Fig. 342.** It is most conveniently used by filling a glass jar, preferably similar to the one in the figure, with the liquid to be tested and then inserting the hydrometer. The reading on the scale of this instrument which is at the level of the surface of the liquid is the specific gravity. The hydrometer shown has a thermometer combined with it, so that the temperature of the liquid can be read on the scale of the upper stem. The "surface of the liquid" is here understood to mean the surface of the main body of the liquid and not the level of the ring around the instrument due to capillarity. Hydrometers are made with two standard scales. One is the ordinary **specific gravity scale** graduated to correspond to the determinations of specific gravity as defined for determinations with the specific gravity bottle; that is, using always the ratio of the weight of the liquid to the weight of an equal volume of water. The other is an arbitrary one known as **Baumé's**

and is much used by trades people. For short, it is often called the " gravity " scale. The following condensed table will be found convenient for converting one scale to the other. As a " rule of thumb " key to the two scales, it will be found useful to remember that 70 in the Baumé scale is approximately 0.70 in specific gravity.

Baumé.	Specific Gravity.	Baumé.	Specific Gravity.
10	1.0000	60	0.7368
20	0.9333	65	0.7179
30	0.8750	70	0.7000
40	0.8235	75	0.6829
50	0.7777	80	0.6666
55	0.7567	85	0.6511

Sometimes engineers will find it necessary to determine the specific gravity of liquids when suitable instruments are not available. Fig. 344 shows a device which can be conveniently used in such cases. It consists of two U-tubes with one of the ends of each connected together by rubber tubing. Each U-tube is provided with the usual scale for observing the difference in level of the liquid in the tubes. One of these tubes is to be filled with clean distilled water (condensed steam) and the other with the liquid to be tested. When a slight pressure is produced in the tubes **A** and **B**, as for example by blowing with the mouth, the differences in level of the liquids in the two tubes is to be observed. The difference in level will be greatest, of course, in the tube having the lighter liquid. The ratio of the difference in level in the U-tube containing water to the difference in the level in the U-tube containing the liquid being tested is the specific gravity required. In the figure shown if the U-tube **A** contains distilled water and **B** the liquid being tested, and the water is displaced **a** inches and the other liquid **b** inches, then the specific gravity is **a** ÷ **b**. The advantage of this sort of device is that it can be made up in any place where merely glass and rubber tubing are available.

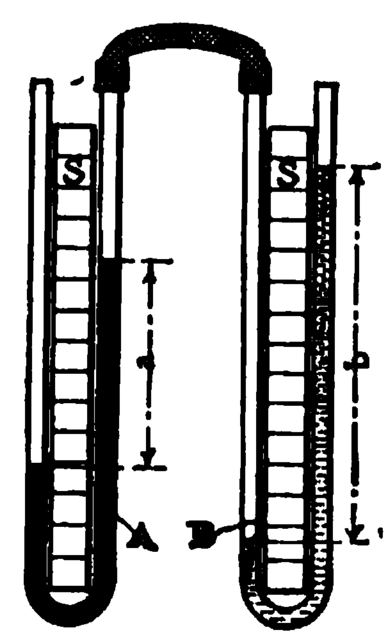


FIG. 344. — Simple Device for Determining Specific Gravity.

Viscosity Test. Every experienced user of oils knows how to determine roughly the relative fluidity or what he calls the " body " of an oil by rubbing it between his thumb and fingers. But such a test gives us no definite standard of comparison. In technical language this **relative fluidity** or " body " is called **viscosity.** It is generally accepted that the viscosity determination is a most important test of the lubricating

qualities of an oil. In general the higher the viscosity the greater the lubricating value, but it should be observed, however, that care must be taken in interpreting the results of such tests because it does not follow, particularly in the case of light, high speed machinery, that a highly viscous lubricant like castor oil would lubricate a spindle for such machinery better than a "thin" sperm oil. In other words as regards viscosity the lubricant must be selected with due consideration to the kind of work to be done.[1]

A lubricant is applied to the bearings of machinery to keep the metallic surfaces moving over each other without coming into direct contact. Unless the surfaces are kept apart by some suitable fluid substance the irregularities which exist on all surfaces, no matter how carefully they are made, will tend to interlock, and the friction caused by forcing or pushing them apart will generate heat. It is the function of a lubricant to flow between the closely fitting surfaces of a bearing and maintain always a thin fluid film to act as a cushion separating the solid particles of the bearing surfaces as well as to take up and carry away either by a direct flow or by vaporization a large part of the heat generated by friction.

Viscosimeters are instruments designed to determine the viscosity of oils. There is no generally accepted standard for such tests as various types of instruments are used; and the results with different instruments will often vary considerably. Unless the names of the designer and maker of the instrument used and the amount and temperature of oil used are stated results are almost meaningless.

It is generally considered necessary to make determinations of the viscosity of the so-called engine, dynamo, and machine oils at 70 degrees Fahrenheit, which is considered the normal temperature in engine and dynamo rooms and in workshops, and again at about 120 degrees which is the temperature of a bearing that is running quite warm. Very often it will be observed that of two samples of engine oil showing the same

[1] Further in regard to the purely physical properties of oils the viscosity determination is a test for the combined effects of both **cohesion** and **adhesion**. A good lubricant must have both of these properties to possess a satisfactory "body." **Cohesion,** as it were, binds together the particles of oil. The more cohesion there is between the particles the greater the pressure or force they will resist before separating. High cohesive value is therefore essential in lubricants intended for heavy machinery. **Adhesion** is the running-mate of cohesion. It is the property of particles of one body adhering to particles of another body of different constitution. An oil with satisfactory adhesive qualities will stick or **adhere** closely to the surface of metallic bearings, and thus assist the cohesive qualities by helping to resist the separation of the particles of the lubricant under pressure. A fluid having high cohesive properties and little adhesion would be a poor lubricant, as, for example, mercury; and conversely great adhesive qualities without those of cohesion as in the case of water, alcohol, etc., are equally unsatisfactory. A good lubricant must have these two qualities jointly to make it viscous.

viscosity as 70 degrees, one will test considerably higher at 120 degrees than the other. Cylinder and gas-engine oils should be tested for viscosity at 100, 200 and 300 degrees Fahrenheit; or if the cylinder oil is used in engines using superheated steam or the gas-engine oil is for internal combustion engines using extremely high compression, even higher limits of temperatures should be used in testing.[1] The best lubricant for a bearing under normal conditions may not do so well when heating begins. An oil which under ordinary conditions would be much too viscous for light, high-speed machinery so as to be very wasteful of power is often most serviceable in a hot bearing when the engine oil ordinarily used vaporizes as soon as it enters and therefore serves no useful purpose. Heavy cylinder oil, however, since it vaporizes at a much higher temperature than the engine oil will be heated only enough to make it thin and limpid without burning and will flow freely between the surfaces of the bearing, keeping the surfaces apart and at the same time conveying away much of the heat generated.

Types of Viscosimeters. One of the simplest instruments and probably the one most used in America for determining the viscosity of oils is shown in **Fig. 345**. It is known as Scott's and consists of a metal cup with an orifice at the bottom and is surrounded by an outer vessel also made of metal which for low-temperature determinations is filled with water. For tests at temperatures above the boiling-point of water the vessel is to be filled with some kind of oil vaporizing at a temperature higher than that required for the tests to be made. The outer vessel is not circular but has a side extension shown on the right-hand side in the figure, under which a gas burner or an oil lamp can be placed for heating the liquid surrounding the inner cup containing the oil to be tested. By this means the oil in the cup can be heated uniformly to any desired temperature. The orifice is kept closed by means of a ball valve on a rod which extends up through the cover. The valve is lifted to allow the flow of oil through the orifice by pressing down with the finger on the end of the lever **A**. The handle **H** is provided for lifting out the oil cup

FIG. 345. — Simple Type of Viscosimeter.

[1] When oils are to be tested at temperatures above about 200 deg. Fahrenheit usually the sample tested is heated by means of a bath of engine oil instead of water in the outer receptacle of a viscosimeter, particularly those of the "orifice" type where the direct application of a flame is impracticable.

so that it can be readily cleaned. A thermometer T registers the temperature of the oil. For draining the water or oil in the outer vessel a cock C is provided, and a glass cylindrical flask accurately graduated in cubic centimeters is supplied for measuring the discharge from the orifice.

To operate this apparatus first fill the oil cup with about 200 cubic centimeters. In this apparatus it is necessary to start always with exactly the same amount of oil in the cup; otherwise there will be variable heads (pressures) on the orifice for different tests, introducing corresponding errors in the flow of the oil to be tested. Fill the outer vessel with water and heat the latter until the oil is at the required temperature for the test. Maintain this temperature constant for two or three minutes and then place a 50-cubic-centimeter flask under the orifice and by removing the ball valve start the flow of oil observing the time as accurately as possible, preferably to the fraction of a second. When the level of the oil in the flask has reached the 50-cubic-centimeter mark close the ball valve and again observe the time. The number of seconds required for 50 cubics centimeters of a liquid to flow through the orifice is called the **time viscosity.** Usually instruments of this kind have the **water viscosity** marked on the name-plate. This is the number of seconds required for 50 cubic centimeters of distilled water at **60 degrees Fahrenheit** to be discharged from the orifice. The water viscosity should be checked from time to time because there is sometimes a small accumulation of grease in the orifice which may remain unobserved and would reduce the actual or effective size of the orifice. Some basis for comparison of determinations of viscosity made by the various types of viscosimeters in which the discharge from an orifice is applied can be obtained by calculating what is called the **specific viscosity.** This is the time viscosity divided by the water viscosity. Thus if for a given apparatus the time viscosity is 120 seconds and the water viscosity is 10, then the specific viscosity is 12.[1]

Redwood's and **Engler's** viscosimeters are practically the same as the one described (Scott's). Redwood's is largely used in England and Engler's is the one officially adopted by the German government. None of the types described have orifices of exactly the same size.

Flash-point Testers. The instrument which has been shown by elaborate tests to be by far the most accurate and reliable is known as

[1] Sometimes when oils of very different specific gravities are to be compared a correction is applied to offset the difference in flow through the orifice, causing the heavier oil to flow faster than a lighter one. Thus we obtain the **gravimetric viscosity** for comparison which is determined by dividing the specific viscosity by the specific gravity. For nearly all practical work the specific viscosity gives a sufficiently good basis for comparison of oils.

the **Abel-Pensky** tester.[1] An improved model as developed by the U. S. Bureau of Mines is shown in **Fig. 346.** It consists of a central oil cup **A** to hold the sample to be tested, **C** a water-bath heated by the Bunsen burner **B,** an overflow **D** for oil expanding by heat, **E** an overflow cup, **S** a stirrer driven from the pulley **P** at the top, T_1 a thermometer for determining the temperature of the oil and T_2 for the temperature of the water-bath, **G** a small gas flame mechanically operated to expose the vapor from the oil cup for exactly one second, and **H** an ivory head with which to judge the standard size of the test flame. This apparatus is considerably more complicated than the apparatus ordinarily used for flash tests. Complications come mostly in providing a stirring device and a mechanical device for applying the test flame. The top of the oil cup is closed except for a small aperture which is opened when the test flame is applied.

Duration of the exposure to the test flame is important. Frequent opening of the aperture to expose the test flame permits frequent escape of the vapors and raises the flash point. Exposing the flame at each degree Fahrenheit rise in temperature is recommended.

The testing or exposure of the test flame should begin at least ten degrees Fahrenheit below the estimated flash point.

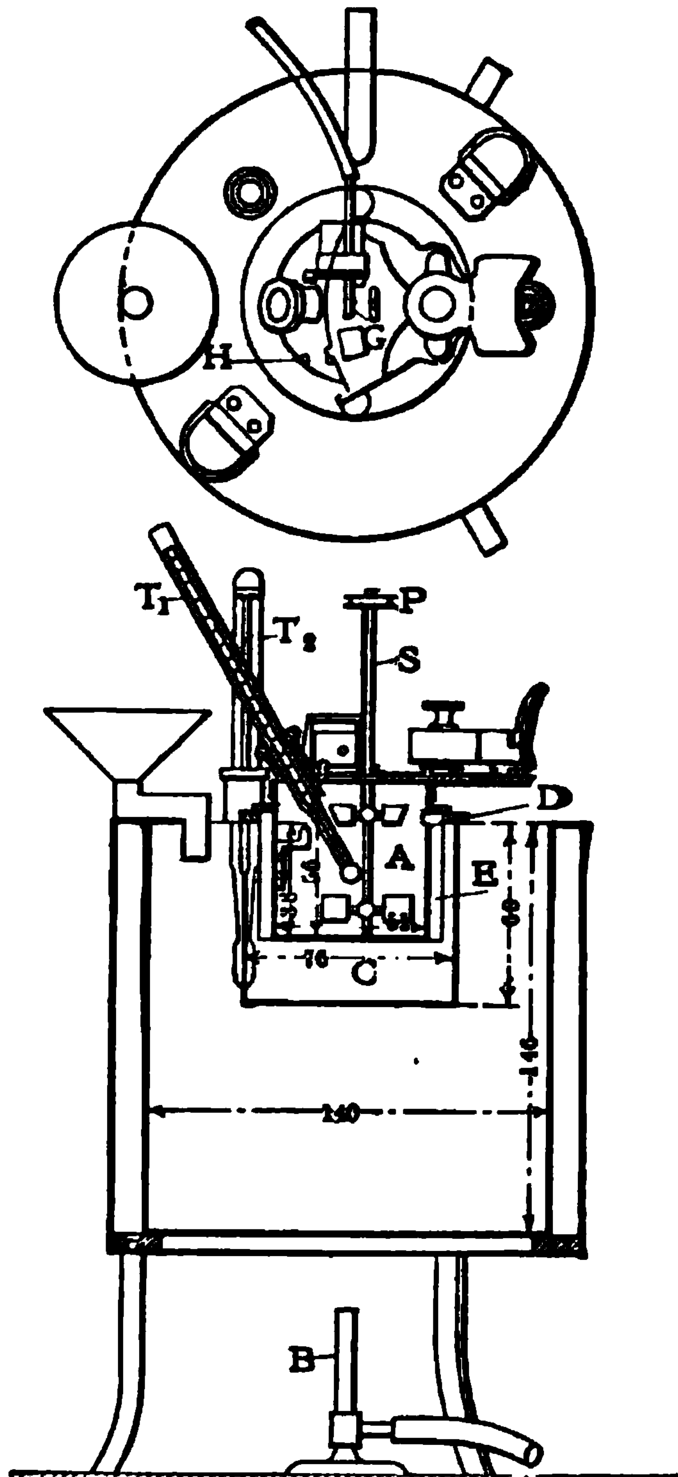


Fig. 346. — U. S. Bureau of Mines Type of Viscosimeter.

The differences of temperature at different points within the oil in the ordinary **unstirred cup** may be as much as from 5 to 10 degrees Fahrenheit. Oil is hottest along the bottom and sides of the cup. The heated oil rises to the surface along the sides, then flows down the center. Therefore the oil and vapors should be stirred during the test.

A coal-gas test flame gives a flash point nearly a degree Fahrenheit lower than an oil test flame. An electric test spark gives a variable but usually a lower result than a gas flame. Since it is not possible to control the intensity and duration of the electric spark an electric test spark should not be used.

[1] For a more complete description see Technical Paper No. 49, U.S. Bureau of Mines, Washington, D. C.

Burning point of an oil is the temperature at which the oil ignites and continues to burn in the cup with the cover removed. This will be usually 5 to 20 degrees Fahrenheit higher than the flash point. The difference is greater for a mixture of light and heavy oil than for a normally refined petroleum product.

The sample and the oil cup must first be brought to a temperature of about 20 deg. F. below the approximate flash point of the oil, by standing the cup in an ice mixture or warming the sample as may be found necessary before the cup is adjusted in the bath. The lower edge of the overflow aperture **D** is greased on its outer side to induce ready overflow when the oil expands. The clean, dry cup is then placed in the bath, the sample is run into the cup with the aid of a glass pipette until the filling point just disappears under the surface as seen by light reflected from the surface of the oil. Care must be taken not to splash the oil on the sides of the cup and not to have froth formed on the oil. All bubbles on the surface of the oil must be pricked with a heated wire. In case too much oil has been accidently run into the cup, the cup must be emptied, washed clean with a good solvent, wiped dry, and a fresh filling made. After the filling is correctly done, adjust the cover and thermometer immediately, light the test flame, and adjust it to the size of the ivory bead on the cover, that is, so that it will burn 0.1 cubic foot of coal gas per hour. Then light the gas flame below the bath and adjust to such a height, as determined by preliminary tests, that the temperature of the oil will rise at the rate of from 4 to 5 deg. F. per minute. For oils flashing above 140 deg. F. a bath of cylinder oil flashing above 300 deg. F. is used instead of water.

Allow the apparatus to stand 10 minutes, to give time for the oil vapors to accumulate, meanwhile stirring regularly and constantly at one revolution per second, then warm to within 10 deg. F. of the flash point and expose the test flame for exactly one second by means of the mechanism provided on the cover. Continue stirring and making the exposure at each deg. F. rise in the temperature until the flash occurs. Particular care must be taken that the cup is not subjected to drafts during the test and that the breathing of the operator is not allowed to interfere, particularly at the moment the test flame is exposed to the vapors. It will be noted that at about 10 deg. F. below the flash point, the test flame, as it is exposed to the vapors, will be surrounded by a pale blue halo, which gradually increases in intensity until a sudden inflammation or gentle explosion of the vapors or the "flash" occurs. The temperature at which this occurs, as registered by the thermometer in the oil, is the flash point. With fuel oil residues or with poorly refined oils from which a small quantity of low-flashing fractions is continually being liberated, this halo may first appear as much as 50 deg. F. below the flash point.

The test should always be repeated with a fresh sample. The first sample should be thrown away, the cup washed with gasoline, wiped dry, and the cup and water bath cooled to the proper temperature.

This instrument with slight variations in dimensions is the official standard in Great Britain, France, Germany, Austria, Denmark, Holland, Belgium, Russia, Italy, Norway, Sweden, Japan and is coming into rapidly extended use in the United States where it has been recommended by government experts for all interstate and foreign trades.

Doubtless in some places the cost of the Abel-Pensky tester prohibits its use. When a high degree of accuracy is not required a simpler type of instrument may be used, observing the precautions as stated. The open cup should be used for all oils flashing above 250 deg. F. For many purposes the flash point and burning points may be determined with sufficient accuracy with an open cup tester used in the following manner.

A glass beaker or metal cup of approximately the same dimensions as the Abel-Pensky and Pensky-Martens testers — that is, 5.0 cm. in diameter and 5.5 cm. in depth — is filled to within 1.8 cm. of its upper rim with the oil to be tested. In testing lamp oils the cup is supported in a water bath by a flange at its upper edge, and in testing oils with higher flash points, the cup is supported on a sand bath. A thermometer is supported with its bulb immersed in the oil. A test-flame burner is made by drawing a piece of hard glass tubing to a capillary about 1 mm. in diameter, to which gas is supplied. The flame is adjusted to 3 mm. cross section. The bath is provided with an ordinary Bunsen burner. The method of testing is as nearly as possible like that in official tests with the Abel-Pensky tester.

Open-cup testers such as the Cleveland and the Tagliabur have been shown by the Bureau of Mines to give flash-points of lamp oils 8 to 10 deg. F. higher than the Abel-Pensky instrument described here, and for heavy lubricating oils the difference may be as much as 100 deg. F. if the special precautions noted here have not been observed. The same investigation shows that the Foster reads about 5 degrees higher than the Tagliabur closed tester, and that the latter is usually about 15 degrees higher than the Elliott tester, and also from 4 to 5 degrees higher than the apparatus described. Obviously the tester giving the lowest tests is the one most nearly correct if the flame exposure is short in duration and is not usually close to the surface of the oil.

Chill Point. To determine the chill point, pour oil of the kind being tested, into a test tube, filling it to a depth of about one-half inch and inserting a low-reading (preferably alcohol filled) thermometer. Place the test tube in a suitable vessel and pack the test tube in a freezing mixture of ice and salt. The vessel should be provided at the bottom with a drain which should be open. After the oil has frozen or congealed, the

chill point is determined by removing the tube from the freezing mixture, invert it at an angle of about 60 degrees, and observe the temperature indicated by the thermometer when the oil starts to run down the sides of the tube. This temperature is the chill point.

Oil Testing Machines. The type of machine generally used in America, for testing an oil to determine its coefficient of friction and its effect in preventing undue heating in a bearing, consists of a horizontal steel shaft supported on two bearings and with one end overhanging its bearing.[1] A pendulum is suspended from the overhanging end by a bronze bearing. This bearing is made in two halves so as to be easily removable for cleaning and scraping. In order to get results at all comparable on different machines the bearing surfaces must be smooth and fit on the shaft as well as they can possibly be made. The pendulum is so arranged by means of an adjusting screw at the bottom that any pressure applied is transmitted in its full amount to both halves of the bearing. The adjusting screw transmits its pressure on the lower half of the bearing by an intermediary spring, thus permitting closer adjustments. A pointer on the spring indicates on a scale attached to the pendulum the total pressure and also the total unit pressure (lbs. per sq. in.) on the two halves of the bearing. A thermometer inserted through a hole in the top of the pendulum, with its bulb resting on the shaft indicates the temperature at the bearing surfaces. An iron ball is provided on some of these devices to partially counterbalance the weight of the pendulum and thus increase its sensitiveness. The deviation of the pendulum from the vertical is obviously proportional to the friction. This deflection is measured by a pointer moving over a circular arc. If the instrument is in proper adjustment the reading on this scale divided by the total pressure exerted on the two halves of the bearing is the value of the coefficient of friction.

Sometimes when the apparatus is to be used at higher pressures than the normal spring permits, the spring can be replaced by a stiffer one, made of heavier wire.[2]

As the mathematical demonstration is usually stated:

$\mathbf{P}$ = total pressure on the two halves of the journal, lbs.

$\mathbf{p}$ = total unit pressure per sq. in. of projected area on the two halves, lbs. per sq. in.

$\mathbf{T}$ = tension in spring, lbs.

$\mathbf{W}$ = gross weight of pendulum, lbs.

$\mathbf{R}$ = effective arm of pendulum, ins.

[1] For a detailed description of slightly different type, (Golden's) see A. L. Westcott in the *Journal of A.S.M.E.*, July, 1913, pages 1143–1167.

[2] The maximum load a spring will carry is proportional to the cube of the diameter of the wire. When making a new spring of heavier wire the outside diameter of the coil must be made the same as of the normal spring, and the unstressed normal length must be the same.

r = radius of journal, ins.
a = angle of deflection of pendulum from the vertical.
F = total force of friction, lbs.
f = coefficient of friction.
l = length of bearing surface.

Since each half is loaded, the total equivalent projected area is $2 \times 2\,rl$.

Total load on journal is $P = 2\,T + W$, and

$$p = \frac{P}{4\,rl} = \frac{2\,T + W}{4\,rl}.$$

Moment of friction is equal to moment of external forces or, since $f = \frac{F}{P}$,

$$Fr = Pfr = (2\,T + W)\,fr = W'R \sin a$$

in the case of the friction deflecting the unbalanced weight W' of the pendulum through the angle a. Then

$$f = \frac{W'R \sin a}{rP}.$$

In any machine of the type described, the term $\frac{W'R}{r}$ is a constant, the scale on the arc indicating deflections of the pendulum is usually graduated to indicate values of $\frac{W'R \sin a}{r}$.

As a rule the total and unit pressures stamped on the scale of the pendulum type of apparatus are not even nearly correct and it becomes necessary to calibrate the spring in the tester, if absolute values are to be calculated. This calibration is accomplished most easily by removing the brasses from the yoke, taking off the lower part of the pendulum which has the thread for tightening up on the spring, and setting up the pendulum with the yoke downward in a testing machine for compression (page 432) and apply increasing loads on the spring and measure its deflection at each point. A stiff bar must be put through the yoke to take the pressure from the spring. This bar can be readily supported on iron blocks so that the pendulum will be free to move up and down with the varying compression on the spring. If all the supporting parts are rigid the deflections of the spring can be measured by the distance between the head and the base of the testing machine.

Steam Engine Lubricators. The proper oiling of an engine is most important. The operation of nearly all types of lubricating devices is easily understood, particularly when operated by the gravity of the oil or by a pump. Another type of lubricator for cylinder lubrication

which is operated by the weight of a column of **condensed steam** is shown in Fig. 348. The pipe C, in which the condensed steam accumulates, must be made at least 2 feet long to give a sufficient head or pressure to feed the oil. The oil reservoir is in the cylindrical vessel below the condenser pipe. This is filled with oil through an opening in the top as the water which has accumulated in the apparatus is drained through the cock D. Water from the condenser pipe C is carried down to the bottom of the oil reservoir by the pipe shown in dotted lines on the left-hand side. Oil, being lighter than the water, remains in the

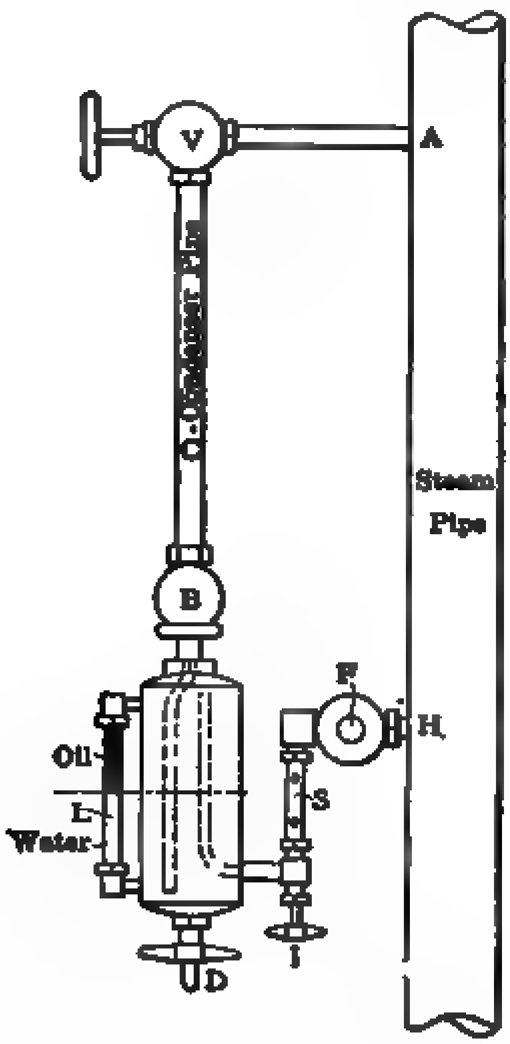


Fig. 348. — Engine Cylinder Lubricator Operated by Pressure of Column of Condensed Steam.

Fig. 349. — "Detroit" Cylinder Lubricator.

upper part of the reservoir and is forced down through the pipe on the right-hand side and through the needle-valve I by which the flow of oil can be regulated so that as small an amount as one drop in two or three minutes passes into the steam pipe at H to mix with the steam going to the engine cylinder. Through the gage glass S the number of drops passing through can be observed. Another gage glass L on the side of the reservoir shows the relative amounts of oil and water. **When an engine is not operating all oil cups and lubricators should be carefully closed.** The feed of oil from this lubricator is stopped by closing

the valves **V** and **F**. A valve is usually provided on the bulb **B**, which should be closed when draining from **D**, so that the water in the condenser pipe **C** will not be lost, and thus prevent the operation of the apparatus until a sufficient amount of condensed steam has accumulated to produce the pressure necessary to force the oil into the steam pipe.

A slight modification of the lubricator described is illustrated in **Fig. 349.** The condensed steam is brought to the bottom of the reservoir through the pipe **P**, which is open at its lower end. On the other hand, the pipe **S** is open at the top and the oil is forced by the pressure due to the head of water in the pipe above (not shown) through the gage glass on the left-hand side, then through the horizontal pipe **T** to which is attached the valve and nipple connected to the steam pipe supplying the engine. In this figure at the top of the reservoir a plug with a nicely finished handle is shown, which is to be removed for filling with oil.

CHAPTER XXII

HYDRAULIC MACHINERY

Tests of Hydraulic Machinery

Tests of Boiler Feed-pumps. In general engineering practice there are two classes of pumps commonly used for "feeding" the water to steam boilers. These are:

(1) Motor- or belt-driven pumps;
(2) Steam pumps.

Motor-driven feed-pumps are more generally used in Europe than in America, but there are, however, many plants in which feed-pumps operated by direct connection to an electric motor or by belting from a line shaft are used. **Fig. 350** shows a good modern example of such a pump. It has three plungers operated from a single shaft with the cranks set 120 degrees apart. The valves are accessible for cleaning or for repairs by removing the plates or "covers" **C, C, C.** The suction pipe is marked **S** and the discharge pipe **D.** An air-chamber **A** is provided to produce, by cushioning air in it, a somewhat more steady flow than would be secured without it. **A relief valve** should be provided on the discharge pipe to act as a safety valve in case the pressure in the line should get so high that the pump itself might be broken.

Fig. 350. — Belt-driven Feed Pump.

The power delivered to a belt-driven pump can usually be conveniently measured with some type of transmission dynamometer (see pages 164–175), while if it is direct-connected to a motor the efficiency of the motor may be obtained by disconnecting it, attaching a Prony brake to the shaft, and measuring the input to the motor with suitable electrical instruments.

The **equivalent work done "on the water"** by the pump is found

by multiplying the total head[1] (suction + discharge) in **feet** by the weight of water lifted (foot-pounds).

The quantity of water delivered can be determined by weighing or by calculating the flow over a weir or from an orifice (see pages 201–207).

Slip is the difference between the volume swept through by the plunger of the pump; or, in general, the piston displacement, and the actual volume of the water pumped at the required head.

In piston pumps, direct-connected to the steam cylinder without a crank, the length of the stroke is usually variable, and some special method must be adopted for such tests to determine the average length of the stroke.

Duty of a pump is usually defined as the number of foot-pounds of work "delivered" by the pump per 1,000,000 B.t.u. supplied. The heat units supplied by the engine are calculated by the A.S.M.E. Rules[2] as the product of the weight of feed-water used by the boiler and the total heat of steam at boiler pressure "reckoned from the temperature of the feed-water." The total heat is to be corrected of course for moisture or superheat.

For a test utilizing a Webber or a similar dynamometer belted to the pump for measuring the power the following form may be used:

TEST OF A BELT-DRIVEN PUMP

(Dynamometer Method)

1. Type of pump............................Made by.....................
2. Diameter of plungers, ins..
3. Length of stroke, ft...
4. Size of suction pipe...
5. Size of delivery pipe..
6. Speed of dynamometer, r.p.m..
7. Speed of pump, r.p.m...

[1] If the **discharge** head is measured by a pressure-gage on the discharge pipe then the equivalent pressure in pounds per square inch corresponding to the difference in level between the surface of the water-supply and the center of the gage must be added to get the total head. (One foot-head of water at about 62 degrees Fahrenheit is equivalent to 0.434 pound per square inch; and conversely, one pound per square inch is equivalent to a head of 2.305 feet of water at the above temperature.)

This is usually stated as the vertical distance between the two gages. A vacuum gage, however, on the suction pipe of a pump indicates the vacuum at the level of the *top of the suction pipe*, and not up to the center of the gage. This was shown by Professor Cooley by attaching a gage by means of a suitable fitting to the suction pipe of a pump so that the gage could be revolved above and below the pipe. It was observed that the reading of the gage remained constant, showing that in a suction pipe of a pump the water does not rise higher than the top of the pipe.

[2] More detailed instructions for steam pumps with steam jackets are given in *Transactions American Society of Mechanical Engineers*, vol. 12, page 530.

8. Dynamometer reading
9. Delivery pressure, lbs. per sq. in.
10. Suction pressure, lbs. per sq., in. or inches vacuum
11. Temperature of water, deg. F.
12. Delivery head in feet of water
13. Suction head in feet of water
14. Total head in feet of water
15. Net weight of water pumped per minute, lbs.
16. Work done by pump. ft.-lbs. per min. (14) × (15)
17. Cubic feet water pumped per minute
18. Plunger displacement, cu. ft. per min
19. Slip, per cent [(18) − (17)] ÷ (18)
20. Net work delivered to pump (by dynamometer) ft.-lbs. per minute.
21. Dynamometer horse power (20) ÷ 33,000
22. Pump horse power (16) ÷ 33,000
23. Mechanical efficiency (22) ÷ (21)
24. Capacity of pump, gallons delivered per 24 hours

A Direct-acting Steam Feed-pump like the one shown in section in **Fig. 351** will be tested in a somewhat different manner, and a different set of observations is required.

In none of the so-called direct-acting steam pumps has a rotary motion been developed by means of which an eccentric can be made to operate

FIG. 351. — Direct-acting Steam Pump.

the valve. It is, therefore, necessary to reverse the piston by an impulse derived from itself at the end of each stroke. This cannot be effected in an ordinary single-valve engine, as the valve would be moved only to the center of its motion, and then the whole machine would stop. To overcome this difficulty a small steam piston is provided to move the main valve of the engine.

In these pumps the lever **A**, which is carried by the piston rod, comes in contact with the tappit when near the end of its motion, and by means of the valve rod **R** moves the small slide valve which operates the supplemental piston. The supplemental piston, carrying with it the main valve **V**, is thus driven over by steam, and the engine is reversed. If, however,

the supplemental piston fails accidentally to be moved, or to be moved with sufficient promptness by steam, the lug on the valve rod engages with it and compels its motion by power derived from the main engine.

Outside-packed steam pumps of the plunger type (Fig. 352) are now very commonly used for supplying boiler feed-water, chiefly because at the pump end the only part subjected to ordinary wear is the packing of

Fig. 352. — Outside Packed Plunger Feed-pump.

the plunger stuffing-boxes. Steam is admitted at A and exhausts at E. The suction pipe is at S and the discharge pipe at D.

Suitable fittings for the attachment of indicators should be provided at both the steam and the water cylinders. If the pump is of the ordinary direct-connected type, without a flywheel, like the one shown in Fig. 351, some provision must be made to make regular observations of

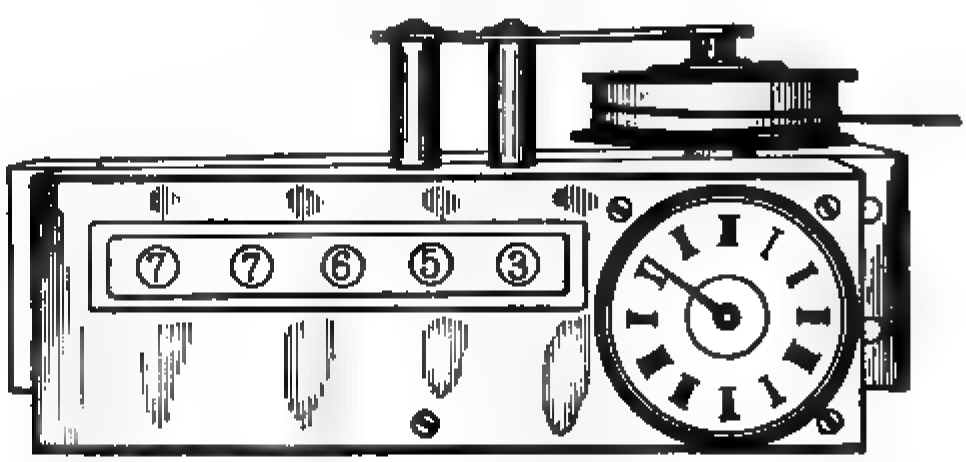


Fig. 353. — Cooley Stroke-measuring Device.

the length of the stroke, as it is scarcely ever constant. One method is to attach a suitable arm to the cross-head H, Fig. 351, with a pencil at the end. Strips of tough paper can then be pasted on a board in such a position that the pencil will trace the lengths of the strokes. By shifting the position of the board every minute or two, records will be obtained from which the average length of the stroke can be estimated with considerable accuracy.

Cooley Stroke Measuring Device.[1] Another method giving still greater accuracy is to use a counting device designed by Professor Cooley, operated by a mechanism similar to that in a clock (**Fig. 353**). A cord from the instrument is attached to the cross-head of the pump and the clock mechanism moved by this cord integrates or sums the lengths of all the strokes. This apparatus has been developed for measuring accurately the length of the stroke of the type of pumps in which steam and water cylinders are direct-connected on the same piston rod, such for example as the ordinary steam feed-pumps. In such pumps it scarcely ever happens that there are two strokes in succession that are of the same length and more or less approximate methods are usually adopted

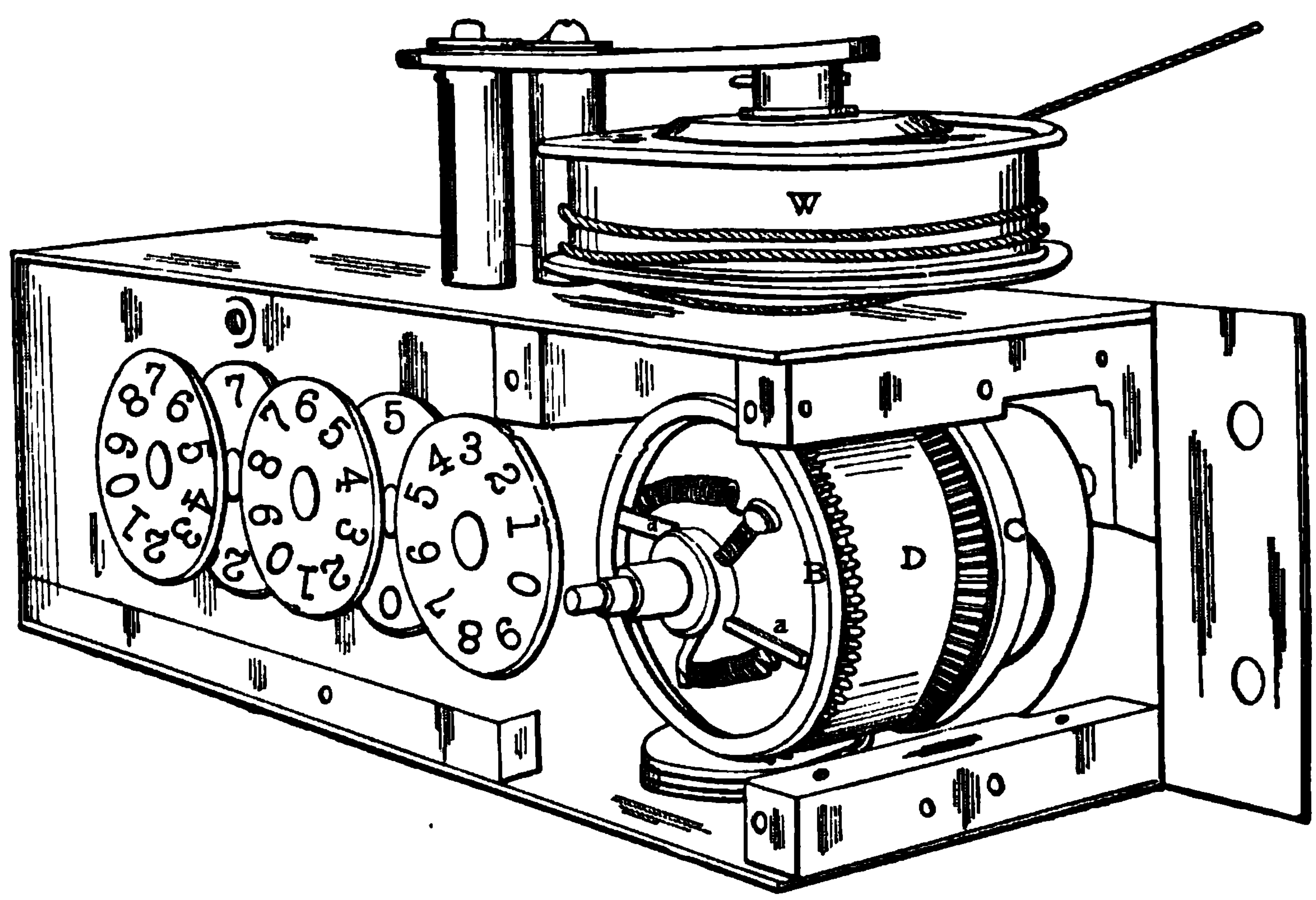


FIG. 354. — Mechanism of Stroke-measuring Device.

for obtaining the average length of the stroke during a test. With the stroke-measuring device referred to above each individual stroke is accurately measured and is added by a counting device to the sum of all the other strokes that have preceded. As this apparatus is used in a test of a variable stroke pump, the reading of the counter to the nearest inch can be recorded at the usual times for observations. The difference between two readings is the total length of all the strokes for the interval between observations. If, then, this difference is divided by the total number of strokes for the same time, the average length of the stroke can be determined accurately. An assembled view of this device is shown in **Fig. 353**, and its mechanism is shown in **Fig. 354**, which it will be observed is the same in principle as the silent ratchet clutches used for the continuous indicator described on page 106. The apparatus

[1] Made by Engineering Shops, Ann Arbor, Mich.

is driven by the cord on the wheel **W**, which moves the ratchet wheels **B** and **C** in the same way as the corresponding parts are moved in the continuous indicator referred to. Numbers on the horizontal plate (Fig. 353) are **feet** and those on the circular dial are **inches.**

RULES FOR CONDUCTING DUTY TRIALS OF STEAM PUMPING MACHINERY.[1] A.S.M.E. CODE OF 1912

Read the general instructions given on pages 258 to 266. Determine the object, take the dimensions, note the physical conditions not only of the pumping machinery but of all parts of the plant concerned, examine for leakages, install the testing appliances, etc., as there pointed out, and prepare for the test accordingly.

In a reciprocating pump determine the quantity of **water leakage** or **slip** past the plungers, and that of the pump valves, if any, as explained on pages 409 and 414.

The apparatus and instruments required for a simple duty trial of pumping machinery, in which the steam consumption is determined by feedwater measurement, are:

(*a*) Tanks and platform scales for weighing water (or water meters calibrated in place).
(*b*) Graduated scales attached to the water glasses of the boilers.
(*c*) Pressure gages, vacuum gages, and thermometers.
(*d*) A steam calorimeter.
(*e*) A barometer.
(*f*) A tachometer or other speed-measuring apparatus.
(*g*) For rotary pumps a weir or other means for measuring the quantity of water pumped.
(*h*) Stroke scales or measuring devices, for direct-acting pumps.

In trials of a reciprocating pumping engine, involving the determination of the complete performance, the weir or other means of measurement noted should be provided, and in addition the following:

(*i*) Steam-engine indicators.
(*j*) A planimeter.

The steam consumed by steam-driven auxiliaries which are required in the operation of the pumping machinery should be included in the total steam from which the heat consumption is calculated, the same as noted in the Steam Engine Code.

In trials for maximum duty care should be taken that no air is snifted into the pump cylinders, causing imperfect filling. In such cases, and indeed in all cases where air is thus admitted in sufficient quantity to

[1] In the case of a pump driven by some other prime mover than a steam engine or steam turbine, the code may be modified to suit the particular circumstances.

affect the performance as revealed by indicator diagrams from the water end, the result should be corrected accordingly.

The rules pertaining to the subjects Duration, Starting and Stopping, and Records are identically the same as those given under the respective headings in the Steam Engine Code, pages 294 and 296, and reference may be made to that code for the necessary directions in these particulars. Where the pump-end is of the reciprocating class, the indicator diagrams should be taken not only from the steam cylinders but also from the water cylinders.

Water Horse Power. The water horse power in a reciprocating pump is found by multiplying the net area of the plunger in sq. in. by the total head, which is made up of the pressure shown by the gage on the force main, that on the suction main, and that representing the vertical distance between the centers of the two gages, all expressed in lb. per sq. in.; the length of the stroke in ft.; and the number of single strokes per minute; and dividing the final product by 33,000.

In a rotary pump the water horse power is found by multiplying the weight of water discharged per hour in lbs., as determined by weir or other measurement; by the total head in ft., as determined from the readings of the gage on the force main, that on the suction main, and the vertical distance between the two gages; and the product divided by 1,980,000.[1]

Duty. The duty per million heat units is found by dividing the number of ft.-lbs. of work done during the trial by the total number of heat units consumed; and multiplying the quotient by 1,000,000. The amount of work is found in the case of reciprocating pumps by multiplying the net area of the plunger in sq. in., the total head expressed in lb. per sq. in. (which is made up of the pressure shown by the gage on the force main, that on the suction main, and the vertical distance between the centers of the two gages, all reduced to lbs.), by the length of the stroke in ft., and the total number of single strokes during the trial; finally correcting for the percentage of leakage of the pump. In a rotary pump the work done is found by multiplying the weight of water discharged during the trial, as determined by weir or other measurement, by the total head in ft.

The duty per 1000 lb. of dry steam is found by dividing the ft.-lbs. of work done, as noted, by the total weight of dry steam, and multiplying the quotient by 1000.

Capacity. The capacity in gal. per 24 hrs. for reciprocating pumps is found by multiplying the net area of the plunger by the length of the stroke in ft. (in direct-connecting engines the average length of stroke); then by the number of single strokes per minute; and the product of these three by the constant 74.8; finally correcting for the percentage of leakage of the pump.

Leakage of Pump. The percentage of leakage is the percentage borne by the quantity of leakage found on the leakage trial, to the quantity of water discharged on the duty run determined from plunger displacement.

Friction. The percentage of total friction in a reciprocating pump is the percentage borne by the friction horse power to the indicated horse power of the steam cylinders.

[1] If there is a material difference in velocity of the water at the points where the gages are attached, a correction should be made by the corresponding difference in "velocity head."

Miscellaneous. For the calculation of other results pertaining specially to the performance of the steam end of a reciprocating pump, reference may be made to the Steam Engine Code.

DATA AND RESULTS OF DUTY-TRIAL OF STEAM PUMPING MACHINERY CODE OF 1912

(1) Test of..........................pump, located at........................
to determine..........................conducted by...................
(2) Type of machinery...
(3) Rated capacity in gals. per 24 hrs...
(4) Type of boiler...
(5) Type of auxiliaries...
(6) Dimensions of engine or turbine..
(7) Dimensions of pump...
(8) Dimensions of boilers...
(9) Dimensions of auxiliaries...
(10) Dimensions of condenser...
(11) Date..
(12) Duration..hrs.

Average Pressures and Temperatures

(13) Steam pressure at boiler by gage.............................lbs. per sq. in.
(14) Steam pipe pressure near throttle, by gage.....................lbs. per sq. in.
(15) Barometric pressure of atmosphere in ins. of mercury....................ins.
(16) Pressure in receiver by gage..................................lbs. per sq. in.
(17) Vacuum in condenser in ins. of mercury.....................................ins.
(18) Pressure in force main by gage..............................lbs. per sq. in.
(19) Pressure in suction main by gage...........................lbs. per sq. in.
(20) Vertical distance between centers of two gages...............................ft.
(21) Total head, expressed in lbs. pressure.......................lbs. per sq. in.
(22) Total head, expressed in ft..ft.
(23) Temperature of main supply of feedwater to boilers....................deg. F.
(24) Temperature of additional supplies of feedwater........................deg. F.
(25) Temperature of air in engine room...deg. F.

Total Quantities

(26) Water fed to boilers from main source of supply...........................lbs.
(27) Water fed from additional supplies...lbs.
(28) Total water fed to boilers from all sources..................................lbs.
(29) Moisture in steam or superheating near throttle..............per cent or deg. F.
(30) Factor of correction for quality of steam, dry steam being unity...............
(31) Total dry steam consumed for all purposes...................................lbs.
(32) Total leakage of plungers and valves...gals.
(33) Total number of gallons of water discharged:
(*a*) By plunger displacement, uncorrected.............................gals.
(*b*) By plunger displacement, corrected for leakage....................gals.
(*c*) By weir or other measurement of discharge.........................gals.
(34) Total weight of water discharged:
(*a*) By plunger displacement, corrected..................................lbs.
(*b*) By measurement of discharge...lbs.

Hourly Quantities

(35) Water fed from main source of supply . . . . lbs.
(36) Water fed from additional supplies . . . . lbs.
(37) Total water fed to boilers per hour . . . . lbs.
(38) Total dry steam consumed per hour . . . . lbs.
(39) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant . . . . lbs.
(40) Net dry steam consumed per hour . . . . lbs.
(41) Dry steam consumed per hour:
  (*a*) By main engine or turbine . . . . lbs.
  (*b*) By auxiliaries . . . . lbs.
(42) Water discharged per hour:
  (*a*) By plunger displacement, corrected . . . . lbs.
  (*b*) By measurement of discharge . . . . lbs.

Heat Data

(43) Heat units per lb. of dry steam based on temperature, Line 23 . . . . B.t.u.
(44) Heat units per lb. of dry steam based on temperature, Line 24 . . . . B.t.u.
(45) Heat units consumed per hour based on main supply of feed . . . . B.t.u.
(46) Heat units consumed per hour based on additional supplies of feed . . . . B.t.u.
(47) Total heat units consumed per hour for all purposes . . . . B.t.u.
(48) Loss of heat per hour due to leakage of plant, drips, etc . . . . B.t.u.
(49) Net heat units consumed per hour . . . . B.t.u.
(50) Heat units consumed per hour:
  (*a*) By engine or turbine alone . . . . B.t.u.
  (*b*) By auxiliaries . . . . B.t.u.

Indicator Diagrams

(51) Mean effective pressure . . . . lbs. per sq. in.

Speed and Stroke

(52) Revolutions per minute . . . .
(53) Number of single strokes per minute . . . .
(54) Average length of stroke . . . . ft.

Power

(55) Indicated horse power developed:
  1st cylinder . . . . i.h.p.
  2d cylinder . . . . i.h.p.
  Whole engine . . . . i.h.p.
(56) Water horse power . . . . h.p.
(57) Friction h.p. (Line 55 − Line 56) . . . . fr. h.p.
(58) Percentage of i.h.p. lost in friction . . . . per cent

Duty

(59) Duty per million heat units . . . . ft.-lbs.
(60) Duty per thousand lb. of steam . . . . ft.-lbs.

Work Done per Heat Unit

(61) Ft.-lbs. of work per B.t.u. (Line 59 ÷ 1,000,000) . . . . ft.-lbs.

Capacity

(62) Number of gals. of water pumped in 24 hrs.:
- (a) By plunger displacement, corrected for leakage....................gals.
- (b) By weir or other measurement......................................gals.

(63) Gallons of water per minute:
- (a) By plunger displacement, corrected for leakage....................gals.
- (b) By weir or other measurement......................................gals.

Economy Results, Steam End

(64) Heat units consumed per i.h.p. per hour:[1]
- (a) By engine and auxiliaries...B.t.u.
- (b) By engine alone...B.t.u.
- (c) By auxiliaries..B.t.u.

(65) Dry steam consumed per i.h.p. per hour:
- (a) By engine and auxiliaries...lbs.
- (b) By engine alone...lbs.
- (c) By auxiliaries..lbs.

RULES FOR CONDUCTING TESTS OF COMPLETE STEAM PUMPING MACHINERY PLANTS

Object and Preparations

This code applies to a commercial test of a complete steam pumping machinery plant, having for an object the determination of the fuel cost of pumping a given quantity of water for a day's run of 24 hours. For tests of the component parts of the plant, rules may be found in the respective codes applying thereto.

Determine the character of the fuel to conform with the object in view. To obtain maximum economy or capacity, the fuel should be some kind of coal that is regarded as a standard, as noted in the Boiler Code, page 269.

The duration of a test of a complete pumping machinery plant should be not less than one day of 24 hours.

In cases where the machinery is in operation only a part of the calender day, the coal consumption used in computing the results should be that of the entire 24 hours.

The methods of starting and stopping a test of a complete pumping plant, sampling and drying coal, determining ashes and refuse, and ascertaining the calorific value and chemical analysis of the coal are the same as those described in the Code for Complete Steam Power Plants, pages 231 and 337.

For the calculation of water horse power, duty, capacity, and leakage of pump reference may be made to the explanations given on pages 413 and 414 of the Pumping Engine Code.

[1] The i.h.p. on which the economy results are based is that of the main engine given in Line 54.

DATA AND RESULTS OF TEST OF STEAM PUMPING MACHINERY PLANT CODE OF 1912

(1) Test of.................................plant located at...............
to determine.......................conducted by.................
(2) Type of machinery..
(3) Rated capacity in gallons per 24 hrs...................................
(4) Type of boilers..
(5) Type of auxiliaries..
(6) Dimensions of machinery..
(7) Dimensions of boilers..
(8) Dimensions of auxiliaries...
(9) Dimensions of condenser...
(10) Date...
(11) Duration..hrs
(12) Length of time machinery was in motion with throttle open...............hrs.
(13) Length of time machinery was running at normal speed.................hrs.

Average Pressures and Temperatures

(14) Steam pressure at boiler by gage.............................lbs. per sq. in.
(15) Pressure in force main by gage...............................lbs. per. sq. in.
(16) Pressure in suction main by gage..............................ins. mercury
(17) Vertical distance between centers of two gages..............................ft.
(18) Total head expressed in lbs. pressure.........................lbs. per. sq. in.
(19) Total head expressed in ft..ft.

Speed and Stroke

(20) Revolutions per minute...
(21) Number of single strokes per minute......................................
(22) Average length of stroke...ft.

Total Quantities

(23) Total coal as fired...lbs.
(24) Moisture in coal...per cent
(25) Total dry coal consumed..lbs.
(26) Ash and refuse...lbs
(27) Percentage of ash and refuse to dry coal..........................per cent
(28) Calorific value per lb. of dry coal by calorimeter test..................B.t.u
(29) Cost of coal per ton of .... lbs...dollars
(30) Total leakage of plungers and valves.......................................gals
(31) Total number of gals. of water discharged:
(*a*) By plunger displacement, uncorrected..............................gals.
(*b*) By plunger displacement, corrected for leakage..................gals.
(*c*) By weir or other measurement..gals.

Hourly Quantities

(32) Dry coal consumed per hour, based on duration of running period (Line 25 ÷ Line 12)..lbs.

POWER

(33) Water horse power...h.p.

ECONOMY RESULTS

(34) Dry coal consumed per water h.p. per hour..............................lbs.
(35) Cost of coal per water h.p. per hour..................................cents
(36) Duty per 100 lbs. of dry coal...

CAPACITY

(37) Number of gals. of water pumped in 24 hrs.:
 (a) By plunger displacement, corrected for leakage.....................gals.
 (b) By weir or other measurement.......................................gals.
(38) Number of gals. of water pumped per minute:
 (a) By plunger displacement, corrected for leakage.....................gals.
 (b) By weir or other measurement.......................................gals.

Tests of Centrifugal Pumps. Tests of pumps operating against low heads such as single-stage centrifugal pumps are made in the same way as explained, for the triplex belt-driven feed-pump. It is desired, of course, from the results of the tests to compare the power supplied to the pump with the work done in lifting the water. Power supplied would probably be again measured by some form of transmission dynamometer, and the work done is calculated from the weight of water delivered and the total head against which the pump delivers.[1]

Centrifugal pumps are frequently driven by direct-connected steam turbines. The horse power required to drive the pump is then determined from a speed-power curve similar to the turbine (**Fig. 308**), page 316, obtained usually from a Prony brake test of the turbine. Similarly, if the pump is driven by a variable-speed electric motor, a speed-power curve of the motor can be used. Usually, however, when a constant speed motor is used it is simpler to determine an efficiency curve of the motor for varying power.

FIG. 355. — Bucket of an Impulse Water Wheel.

Tests of Impulse Water Wheels. Impulse wheels used to operate with water under pressure consist usually of a series of buckets attached to

[1] For more detailed testing of centrifugal pumps see "Centrifugal Pumps," by Lowenstein and Crissey (D. Van Nostrand Co., 1911).

Fig. 356. — Typical Impulse Water Wheel.

Fig. 357. — Water Jet Discharging at High Pressure from the Nozzle of an Impulse Wheel.

the periphery of a disk or wheel. The buckets are usually divided by a central rib so that two " pockets " are formed (**Fig. 355**). The curves for each of the divisions of the bucket are designed to turn the direction of the impinging steam without shock. **Fig. 356** shows a typical impulse wheel. Impulse wheels are designed to operate most efficiently with high heads. It is, therefore, impracticable to measure the head directly in feet, but it is done usually by measuring the pressure near the nozzle **N** with a gage. When the center of the gage is at a higher level than the center of the nozzle discharging on the wheel, then this difference in level must be added to the head calculated from the gage pressure to determine the total head under which the wheel is operating. Power developed is measured usually by a Prony brake connected to the shaft

Fig. 358. — Buckets and Jet of a Pelton Wheel.

S. In all tests where a large quantity of water is used, the temperature of the water should be recorded and the weight corresponding should be used. A view of the jet discharged from one of these nozzles is given in **Fig. 357**. The type of impulse wheel most in use commercially is called the Pelton, of which typical buckets and the engaging jet of water are shown in **Fig. 358**.

Laboratory tests for a **given head** are usually run when varying both the load and speed. Make the first test with the load on the Prony brake as light as possible consistent with fairly steady operation of the wheel, and then take a series of tests increasing the load in increments to reduce the speed about 100 revolutions per minute in each succeeding test. Duration of test at each speed should be from twenty to thirty minutes with observations taken every two minutes. The following form may be used for tests:

TEST OF IMPULSE WHEEL

General Data:

1. Date....................
2. Name of wheel and nominal horse power....................
3. Kind of bucket....................
4. No. of buckets....................
5. Angle of buckets....................
6. Diameter of bucket wheel, inches....................
7. Area of nozzle and delivery pipe, sq. ins....................
8. Coefficient of discharge for type of nozzle....................
9. Diameter of brake wheel, inches....................
10. Length of brake arm, inches....................
11. Tare of brake, lbs....................
12. Duration of test....................
13. Average temperature of water, deg. F....................
14. Average pressure by gage at wheel, lbs. per sq. in....................
15. Average head at wheel in feet[1]....................
16. Quantity of water for total run in pounds....................
17. Quantity of water in pounds per minute....................
18. Cubic feet of water per minute....................
19. Foot-pounds of work per minute calculated from (15) and (17)....................
20. r.p.m....................
21. Net weight on brake, lbs....................
22. Horse power as measured by brake....................
23. Over-all efficiency of motor, per cent (22) ÷ (19) × 33,000....................

Curves. Plot a curve for each head with speed for abscissas and efficiency per cent for ordinates, also curves for the ratio of the velocity of the periphery of the wheel v_p to the theoretical velocity due to the head v_t; that is, v_p/v_t for abscissas and the maximum horse power developed for ordinates.[2]

Tests of Water Turbines. A typical reaction turbine is shown in **Fig. 359.** The power is transmitted by the main shaft and the smaller shaft is used for controlling the gates regulating the quantity of water passing through the wheel. For testing a Prony brake can be placed directly on the vertical shaft. In some respects a rope brake is most suitable, as one end can be attached to a spring balance and the other end can be led over a pulley, and will thus support weights on a vertical hanger. It is preferable to have the lower web of the brake-wheel solid, that is, without arms, so that it will retain the cooling water, which should be arranged to flow into it at the rate required.

Power supplied is determined by the weight of water used and by the head under which the wheel operates. These quantities are deter-

[1] Corrected for vertical distance from the center of the gage to the center of the nozzle.

[2] Plot curves showing effect of head on efficiency if several tests are run at different heads.

mined in the same general way as for a test of an impulse wheel, already described, **except in the case of a reaction turbine,** where the housing or casing in which the wheel is placed is **always completely filled with water.** With this arrangement the turbine receives not only the effect due to the pressure-head, measured from the level in the head-race to the center of the wheel, **but also that due to the suction head,** measured from the center of the wheel to the level in the tailrace. Data can be recorded in a form similar to that for tests on impulse wheels.

Fig. 359. — Typical Reaction Water Turbine.

Curves. Plot a curve for each gate opening at a constant head with speed for abscissas and efficiency per cent for ordinates.

Air Lifts. Pumping water by compressed air has had in recent years extensive application. There are several methods in use, but the most successful is that using air expansively for raising water as shown diagrammatically in Fig. 360, called the "Air lift." It consists of a delivery-pipe D set down into the well and a smaller pipe for compressed air

having a nozzle N at the end and entering the discharge pipe as shown. It is a more usual construction to admit the air to the discharge pipe through holes from an annular chamber encircling it, but the method of operation is practically the same as shown. Water is raised by the buoyancy of the air. Let h_1 be the depth of submersion of the delivery pipe measured to the point where the air enters, and h_2 be the total lift measured from the same point. Pressure of air at the place of entrance must be theoretically equal to the pressure corresponding to the head of mixed water and air above it in **D**. This pressure decreases, however, as the air rises and expands so that at the top of **D** the pressure is little above atmospheric. Work required of the compressor varies with the difference between the depth of submersion of the delivery pipe h_1 compared with the net lift $h_2 - h_1$. The pressure required is then comparatively high and the efficiency low. On the other hand, a very small submergence necessitates a relatively large quantity of air to produce the required velocity, so that again the efficiency is low. Maximum efficiency, usually about 50 per cent, is obtained when the net head $h_2 - h_1$ is from 15 to 30 feet. At 150 feet the efficiency is scarcely ever as much as 20 per cent.

Fig. 360. — Air Lift.

The following quantities should be determined in a test: (1) horse power of air compressor ; (2) volume of free air (see page 376, line 72) compressed; (3) weight of water pumped; (4) net lift $(h_2 - h_1)$; (5) efficiency, which is the ratio of the work equivalent of lifting the water to the work done in compressing the air, both in foot-pound units.

Tests of Hydraulic Rams. A section of a typical hydraulic ram is shown in **Fig. 362.** It consists of an air chamber **H**, to which is connected the discharge pipe **I**. There is a check valve **G** opening into the air chamber from the lower chamber **A** into which water is brought by the pipe **S**. There is a waste valve at **B**. This valve is weighted and opens inward. By means of a nut **J** on the stem of this valve the lift or amount of opening of the valve can be regulated. When water is supplied to the ram, it escapes through the waste valve **B** with a velocity corresponding approximately to that due to the head under which the water is supplied. The effect of this velocity head is to reduce the pressure on the upper side

of the valve so that it becomes unbalanced and closes suddenly. Then the momentum of the column of water in the pipe S becomes sufficient to open the valve G, and will discharge some water into the discharge pipe I against a considerable head. As soon as the pressures become equalized the valve G closes, the waste valve B opens and water from the supply pipe is again "wasted." This alternate action is produced with regularity, and as a result the water in the supply pipe acquires a certain "backward and forward" wave-motion. As the rule is generally stated, the length of the supply pipe leading from the reservoir to the ram must be at least five times the head. This length is necessary to secure some resistance to this "backward and forward" wave-motion. A small air chamber shown at P, with a check-valve C opening inward to supply air,

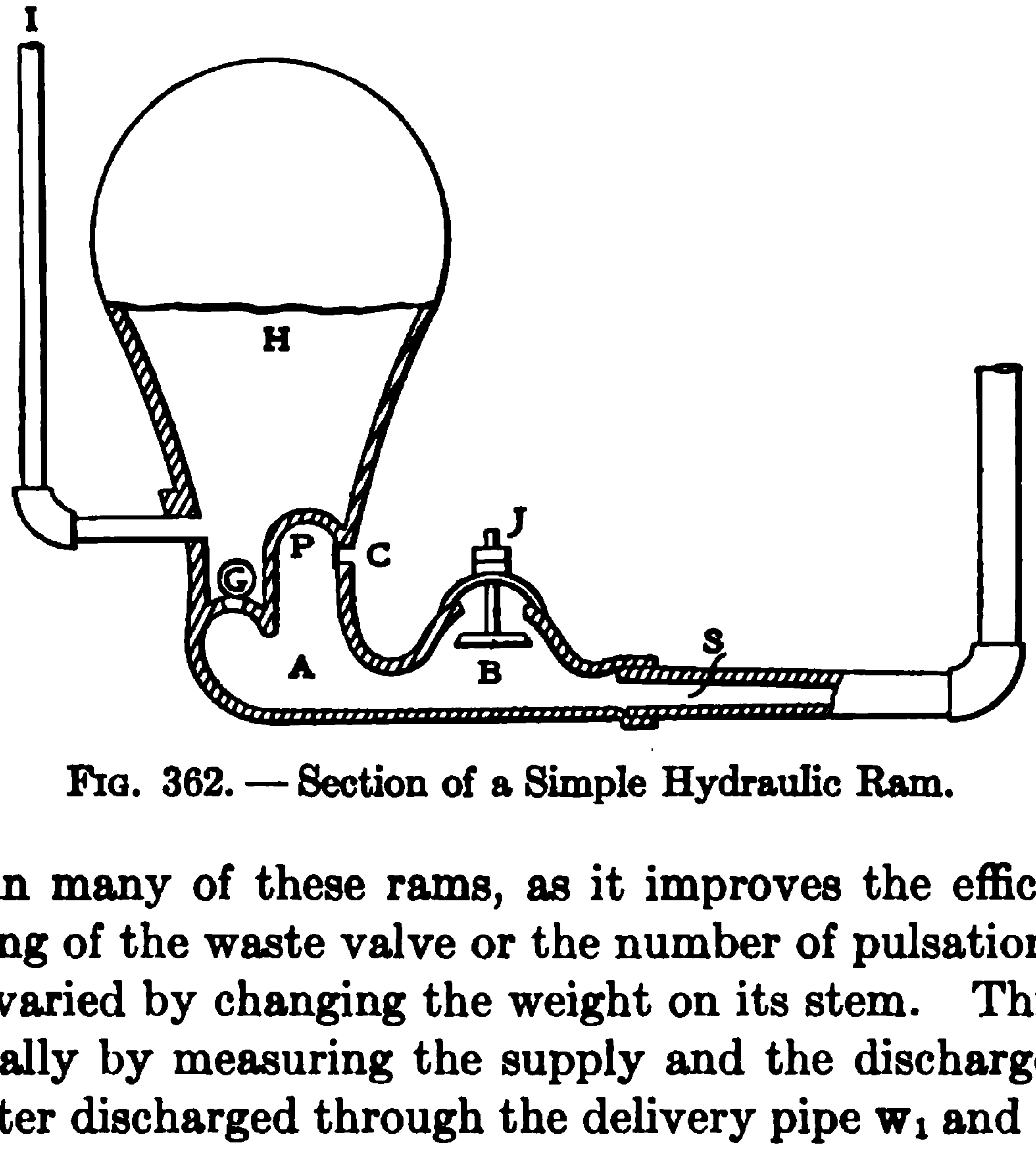


FIG. 362. — Section of a Simple Hydraulic Ram.

is provided in many of these rams, as it improves the efficiency. The rate of opening of the waste valve or the number of pulsations in a given time can be varied by changing the weight on its stem. This apparatus is tested usually by measuring the supply and the discharge heads, the weight of water discharged through the delivery pipe w_1 and that passing through the waste valve w_2 in pounds per minute. Then the available energy in the water is $(w_1 + w_2)h_s$, where h_s is the supply head; and the useful work is w_1h_d, where h_d is the discharge head,[1] then,

$$\text{Efficiency} = \frac{w_1 h_d}{(w_1 + w_2)\, h_s}, \quad \ldots\ldots \quad (131)$$

and the capacity Q in gallons per twenty-four hours is $Q = 1440\, w_1 q$, where q is the fraction of a gallon of water in a pound. Satisfactory runs of twenty minutes' duration can usually be made, each run being made

[1] Both the supply and the discharge heads must be measured, of course, from the same datum or "zero" level.

with a different lift or "stroke" of the waste valve **B.** Observations of the heads should be taken every five minutes if they are variable, and weighing as often as necessary, depending on the size of the tanks used. The effect on the efficiency of increasing the lift or "stroke" of the waste valve from one-eighth inch by increments of one-eighth inch is very interesting.

Curves. Plot curves with length of "stroke" as abscissas and take for ordinates:

(1) Efficiency;

(2) Capacity in gallons per twenty-four hours;

(3) Strokes per minute.

Fig. 363 shows a slightly different form of ram, as made commercially. The principle of operation is, however, the same as the one in **Fig. 362.** Letters used for marking the parts are the same in the two figures.

Tests of Pulsometers. A type of steam pump called a pulsometer is illustrated in section in **Fig. 365.** In the form shown here it consists of

Fig. 363. — Commercial Type of Hydraulic Ram.

Fig. 365. — Steam Pulsometer.

two chambers **AA,** joined by tapering necks into which a ball **C** is fitted so as to move in the direction of the least pressure between seats in these tapering passages. The chambers **AA,** on opposite sides, are connected by means of check or clack valves **EE** with the "induction" chamber **D.** Water is delivered through the passage **H,** which is connected to the chambers through the valves **G.** Between the chambers is also a vacuum

chamber J, connecting them with the "induction" chamber D. Small air valves, moving inward, supply air to the chambers AA by opening when the pressure is less than atmospheric. Its operation is explained briefly as follows: Starting with the left-hand chamber full of water and with a vacuum in the right-hand chamber, this latter chamber will fill with water which by its momentum due to rushing in suddenly, pushes back the valve C toward the left. Now during this time steam has entered the left-hand chamber to the left of the valve C (before it has shifted) and by exerting a pressure on the surface of the water it forces it through the check valve G, first into the delivery passage H and then into the air chamber J. Then the steam in this left-hand chamber condenses in contact with the cold water and forms a vacuum, permitting the repetition of this cycle of events, except that the operations in the two chambers are reversed.

Since all the steam used is condensed and discharged with the water lifted, the analysis of the operations in a pulsometer are similar to those in the familiar types of injectors, except that the steam acts in the pulsometer by pressure instead of by impact as in the injector.

Using the following symbols:

w_s = weight of dry steam, pounds;
w_w = weight of water lifted, pounds;
t_1 = temperature of the water supply, deg. Fahr.;
t_2 = temperature of the water delivered, deg. Fahr.;
r = the latent head of evaporation of the steam in B.t.u.;
t_s = the temperature of the steam, deg. Fahr.;
h_1 = the suction head, feet;
h_2 = the delivery head, feet;
$h_1 + h_2$ = the total head, feet, then

$$w_s(t_s - t_2 + r) = w_w(t_2 - t_1). \quad . \quad . \quad . \quad . \quad . \quad (132)$$

The heat equivalent of the mechanical work done is in B.t.u.,

$$\frac{1}{778}\left(w_w h_1 + (w_s + w_w)\, h_2\right),$$

and the heat expended is in B.t.u.,

$$w_s(t_s - t_2 + r),$$

and

$$\text{Thermal Efficiency} = \frac{w_w h_1 + (w_s + w_w)\, h_2}{778\,(w_s(t_s - t_2 + r))}. \quad . \quad . \quad (133)$$

And if we neglect the work done in lifting the condensed steam,

$$\text{Efficiency} = \frac{h_1 + h_2}{778\,(t_2 - t_1)}. \quad . \quad . \quad . \quad . \quad . \quad (134)$$

Curves. Plot with discharge pressures for abscissas curves with both thermal efficiency and capacity (gallons or pounds of water per twenty-four hours) as ordinates.

Tests of Injectors. The injector is known particularly in stationary service as the device used for pumping water into the boiler when the feed-pump fails. One of the various forms of injectors sold commercially is shown in **Fig. 366.** The **steam supply,** the **suction** or water supply, the **delivery** or discharge, and the **overflow** are marked clearly.

Fig. 366. — Single Tube Steam Injector.

Method of Operating Injectors. The method to be given, although applicable particularly to the ones described, is, however, more or less generally applicable to all makes. Open **wide** both the **steam-** and **water-**supply (suction) valves. Then **close** the water-supply (suction) valve **slowly until the overflow ceases.** Regulate the rate of delivery by closing the water-supply (suction) valve. Before testing an injector or indeed even before trying to operate a new injector, inspect the pipe fittings and particularly the valves on the water-supply pipe to observe whether they are tight. It is not at all unusual to find that the valve is not air-tight, and for this reason it is a very good practice to put always some new wicking in the space for packing around the stem of the valve on the water-supply pipe; and turn up the cap over the packing tightly.

Method of Testing. For the testing of injectors the arrangement of apparatus consists usually of two barrels supported on platform scales, or carefully calibrated tanks fitted with gage glasses. During a test the injector draws water from one barrel or tank and discharges it into the other. A test of an injector must be made, of course, with established conditions; that is, with a flying start. This may be accomplished by having the injector draw water from the supply tank, but discharge water through a by-pass connection on the discharge pipe until the test is to begin. For this preliminary operation of the injector the level in the supply tank can be maintained very closely, at any point marked by

manipulating a "quick-opening" valve. When the test is to begin, close as quickly as possible this valve on the pipe discharging into the supply tank and turn the discharge from the by-pass into the delivery tank. To make this adjustment all the valves to be operated should be of the "quick-opening" type. The pressure against which the injector is to operate is secured by **throttling** the discharge pipe by means of a globe or an angle valve placed between the injector and the by-pass on the discharge pipe. The quick-opening valve would not be satisfactory. The suction head is measured from the middle of the injector to the **average** level of the water in the supply tank. The discharge head is obtained by adding to the head in feet corresponding to the pressure indicated by the gage the distance in feet from the center of the gage to a horizontal line through the middle of the injector. The temperatures of the water in the supply and delivery pipes must be observed. The injector is stopped at the end of the test by closing the steam valve.

The following form, similar to the one used at Purdue University, is very complete. Notes explaining the calculations required are given above:

TEST OF AN INJECTOR

Make of injector...Date.....
Number...
Size of connections: steam............in. dia.; water..........in. dia.; discharge ....in. dia.; area of discharge ($= a$)............................sq. in.
Diameter (minimum) of lifting tube........in.; forcing tube..................in.

a. Duration of test....................................
b. Steam pressure (average) pounds gage, p_s....................................
c. Delivery pressure (average) pounds gage, p_2....................................
d. Maximum pressure against which injector will discharge, p_{max}....................
e. Suction-head (average), feet, h_1....................................
f. Delivery-head (average), feet, h_2....................................
g. Temperature of supply (average) t_1....................................
h. Temperature of delivery (average) t_2....................................
i. Pounds water supplied per hour, w_w....................................
j. Pounds water and steam delivered per hour, w_m....................................
k. Cubic feet of water delivered per hour, Q....................................
l. Wet steam per hour, w_s ($= w_m - w_w$) ....
m. Dry steam per hour, w'_s ($= xw_s$)....................................
n. Water delivered per pound wet steam, pounds ($= w_w \div w_s$)..................
o. Water delivered per pound dry steam, pounds ($= w_w \div w'_s$)..................
p. Velocity of discharge, feet per second, v ($= 144\,Q \div 3600\,a$) ....
q. Energy delivered, raising injection water, B.t.u. per hour......................
r. Energy delivered, heating injection water, B.t.u. per hour......................
s. Energy delivered, velocity of discharge, B.t.u. per hour......................
t. Total energy delivered, B.t.u. per hour....................................
u. Energy supplied, B.t.u. per hour....................................
v. Thermal efficiency as a boiler-feed apparatus....................................
w. Thermal efficiency as a pump....................................

x. Horse power..

y. Dry steam per horse power per hour, pounds..................

The energy of *raising injection water* $= [w_w(h_1 + h_2) + w_s h_2] \div 778$ B.t.u. per hour.

The energy of *heating injection water* $= w_w\,(q_2 - q_1)$ where q_1 and q_2 correspond to t_1 and t_2 B.t.u. per hour.

The *energy of discharge* $= w_m v^2 \div (2\,g \times 778)$ B.t.u. per hour.

The total energy delivered = item q + item r + item s.

The energy supplied $= w_s\,(xr_s + q_s - q_2)_1$ where r_s and q_s correspond to p_s, and q_2 corresponds to t_2. x = quality of steam.

The thermal efficiency as a boiler feed apparatus $= 100 \times \dfrac{\text{item } t}{\text{item } u}$.

The thermal efficiency as a pump $= 100 \times \dfrac{\text{item } q + \text{item } s}{\text{item } u}$.

$$\text{The horse power} = \frac{w_w\,(h_1 + h_2) + w_s h_2}{60 \times 33{,}000} \qquad (135)$$

The dry steam per horse power per hour $= w_s' \div$ item x.

$$\text{The pump duty} = \frac{1{,}000{,}000 + \text{item } p}{\text{item } t}$$

$$= \frac{1{,}000{,}000\,[w_w\,(h_1 + h_2) + w_s h_2]}{778\,w_s\,(xr_s + q_s - q_2)} \qquad (136)$$

The **weight of steam** found by direct weighing may be **checked,** by calculating (assuming radiation loss negligible) a "heat balance" in which this weight will be the only unknown, thus for this condition,

$$w_s\,(xr_s + q_s - q_2) = \frac{1}{778}\left[w_w\,(h_1 + h_2) + w_s h_2 + (w_w + w_s)\,\frac{v^2}{2\,g}\right] + w_w\,(q_2 - q_1),$$

or, approximately,

$$\mathbf{w}_s = \frac{w_w\left[h_1 + h_2 + 778\,(q_2 - q_1) + \dfrac{v^2}{2\,g}\right]}{778\,(xr_s + q_s - q_2) - h_2} \qquad (137)$$

CHAPTER XXIII

TESTING THE STRENGTH OF MATERIALS

Machines for Testing the Strength of Materials consist, in general, of (1) a **power system** for producing in the specimen tested the required stresses, and (2) a weighing system to determine the amount of force applied. In the usual form of testing machine the load is applied to the specimen through a train of gears and screws operated either by power or by hand, depending, of course, largely on the capacity. The stress is measured by balancing the force exerted on the specimen by a poise adjusted at the end of a system of levers just as weight is determined with a platform scales. The general principle of most machines for testing materials is illustrated in a simple form in the apparatus for calibrating indicator springs, **Fig. 120**, page 116. In this case the power is applied to the hand wheel, which exerts two forces equal but opposite in direction, one compressing the spring in the indicator and the other pressing on the platform of the scales.

A diagrammatic view of a typical machine for testing materials in tension and compression is shown in **Fig. 370.** It consists essentially of a table **T,** to which the upper " head " **A** is rigidly attached by means of the vertical bars **DD.** Heavy vertical screws **SS,** carrying the lower cross-head **B,** are moved up or down by the system of gears **GG.** Moving the cross-head **B** downward puts a tensile stress on a test specimen **s** if it is attached firmly to both the upper " head " **A** and to the cross-head **B.** The force applied to the specimen is transmitted by the bars **DD** to the weighing table **T,** which rests on the first weighing lever **M,** having a fulcrum at **F.** The load on the table **T** is applied to the lever **M** at the middle of the table. The long arm of this lever is connected by means of a short link to a second lever **N,** and this again is connected to the short arm of the lever **Q** at the other end of which the weighing poise **P** is to be balanced. The position of the poise on this last lever (scale beam) indicates the force applied to the specimen **s.**

Fig. 371 is a remarkably good illustration for showing the parts of a standard testing machine and for explaining its operation. Vertical screws **SS,** connected by gearing to the power system, move the cross-head **B** up or down according to the direction of motion. The speed of these screws is controlled and their motion reversed by manipulation of the levers marked l_1, l_2, and l_3. The vertical columns supporting the upper " head " **A** are bolted to the table **T,** which rests on the system

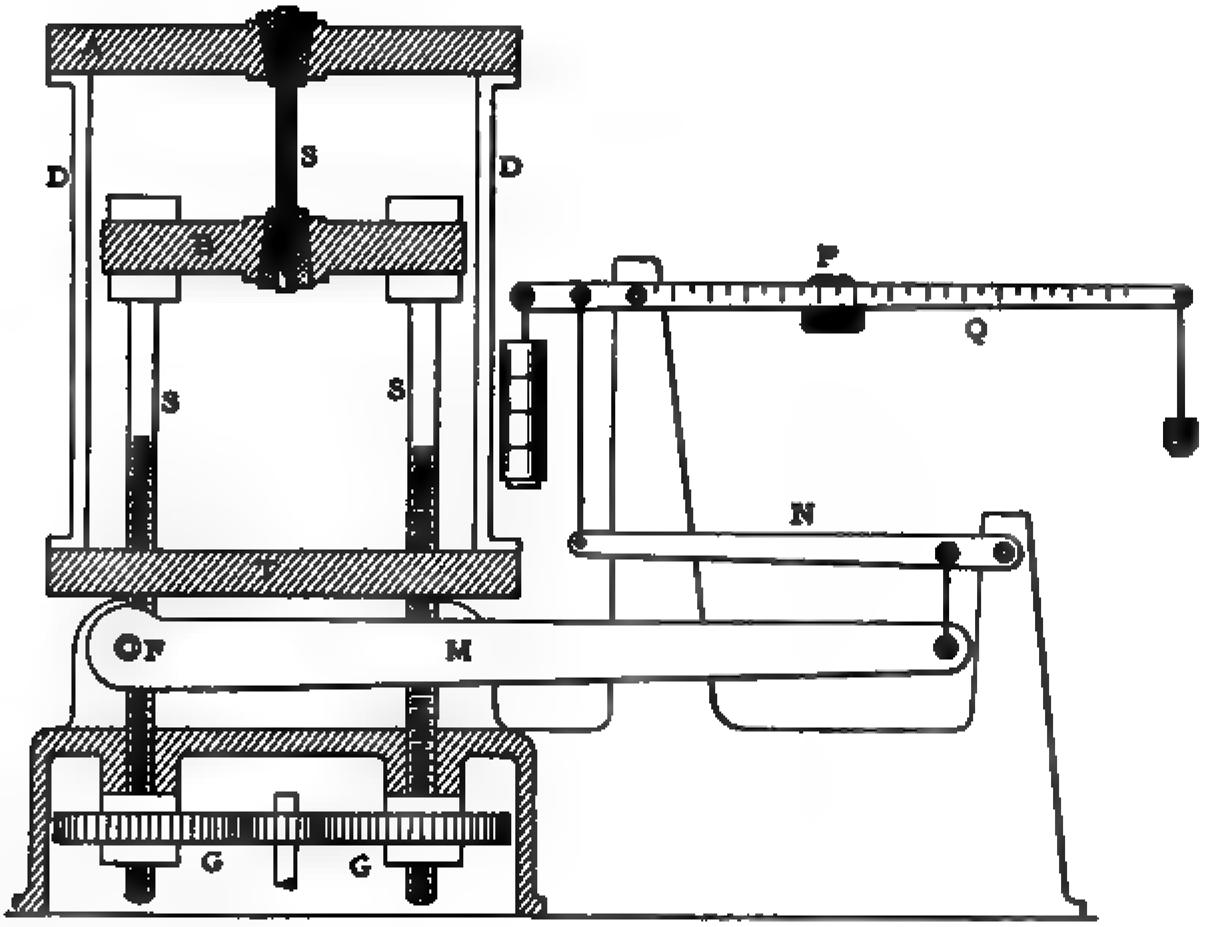


FIG. 370. — Diagram of a Simple Machine for Testing the Strength of Materials.

A

FIG. 371. — Standard Testing Machine.

of levers **M, N, O** and **Q.** The poise **P** is moved on the lever or scale-beam **Q** by means of a cord connected to the hand wheel **W.** The levers are balanced "to zero" by means of the counterpoise **C.** To prevent the sudden jarring of the machine when the load is released by the breaking of a specimen, vertical rods fastened to the base pass up loosely through holes in the table **T,** at its four corners, and on their ends large "**check-nuts**" are screwed. When the machine is in use these nuts must be loose, otherwise they will cause a pressure on the table causing the indication of the scale-beam to be greater than the weight due to the load on the specimen.

Small testing machines with a capacity not exceeding 50,000 pounds are made to operate by **hydraulic pressure.** In this type of machine the movable head applying the load to the specimen is moved by the pressure on a piston in a hydraulic cylinder. This hydraulic pressure is produced usually in a small hand-operated pump at the side of the machine. Oil is generally used for the working medium. In order to return the oil to the pump from the cylinder, when the pressure is to be released, a small check valve controlled by a lever or a screw is usually provided. Machines operating hydraulically are not satisfactory for large loads, because the leakage from the cylinder is likely to be excessive.

When a specimen is to be tested in **tension** its upper end is fastened into the wedges or "jaws" in the upper "head" and its lower end is similarly gripped in the lower or movable cross-head. For tests in **compression** the specimen is placed between the movable cross-head and the table. **Transverse loads** can be applied to long wooden or metal beams with the machine shown in **Fig. 371,** by placing the beam between the supports or abutments **UU′**, and applying the load by means of the movable cross-head **B.** Usually a special fitting with a blunt but "definable" edge to localize the load is inserted into the cross-head **B** for such tests.

Extensometers. Some of the physical properties of materials are determined by the rate of deformation of the specimen as the stress is applied. To measure the deformation some very accurate instruments have been devised, one of which is shown in **Fig. 372.** It consists essentially of a pair of clamps **CC′**, fitted with sharp-pointed thumb-screws for attaching them to the specimen SS. Two rods **B** and **B′**, fitted to the upper clamp on opposite sides of the specimen, are provided at their lower ends with adjustable points to be screwed up or down by means of the small milled wheels **W** and **W′**. Opposite these rods and fastened to the lower clamp are two micrometer screws, usually graduated to **ten-thousandths of an inch,** for measuring the elongation of the specimen. Electrical connections are made, as shown, with a

battery and a bell. As the specimen stretches out the contact points at P and P′ are moved apart and the distance the micrometer screws are raised measures the elongation. With the help of the bell it is possible to make, for all observations, uniformly light contacts.

Fig. 372. — Extensometer.

Deflectometer. A very simple device for measuring the deflection of beams is shown in **Fig. 373**, consisting of a plate P supported upon a steel bar attached to the end supports UU′. Deflections can be measured with this apparatus with the aid of ordinary "inside" calipers, micrometer calipers, or with a special deflectometer. This instrument illustrated in **Fig. 374**, is often used to measure the deflection of wooden and metal beams subjected to transverse stress. It can also be used with success to measure the contraction of short specimens in compression.

Physical Properties of Materials Defined. The **elastic limit** is a more or less definite value of the unit stress beyond which, as the stress is increased, the increase in deformation is greater relatively than the increase in stress; and further, at this point, the deformations produced will not disappear entirely when the stress is removed. **Permanent set** or "**set**" is used to represent the lasting deformations produced by stresses greater than the elastic limit.

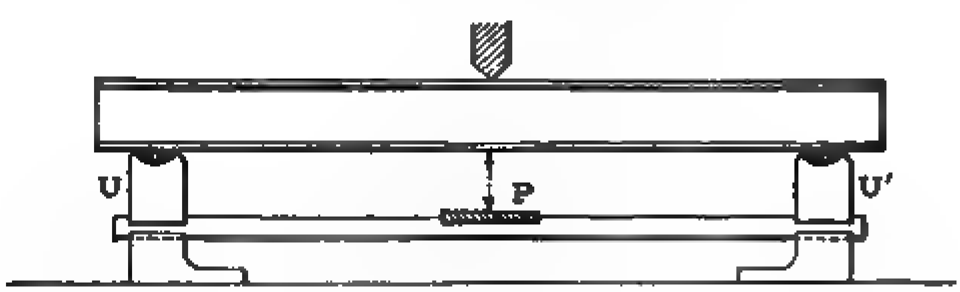


Fig. 373. — Simple Device for Measuring the Deflection of Beams.

Modulus of Elasticity is a term used to express the ratio of the unit stress to the deformation per unit of length[1] accompanying that stress, within the elastic limit. For example, if **f** is the stress in pounds per square inch within the elastic limit and **s** is the accompanying defor-

[1] This unit deformation is often called the unit elongation for slender test-pieces, and more generally the strain.

mation per inch of length in inches, then the modulus of elasticity, **in pounds per square inch, is**

$$\mathbf{E} = \mathbf{f/s}. \qquad (140)$$

The total stress under which a body fails is called its ultimate strength; and the corresponding unit stress is called the ultimate unit strength, or, for short, simply the **ultimate strength.** The ratio of the total elongation of a body to its original length is called the **percentage of elongation.** It is obviously the same as the term unit deformation. For calculations of percentage of elongation measurements are taken, according to convention, between two gage marks usually 8 inches apart. This percentage of elongation is a measure of the **ductility** of the material tested. The **ratio** of the smallest area after rupture to the original area is called the "**reduction of cross-section.**"

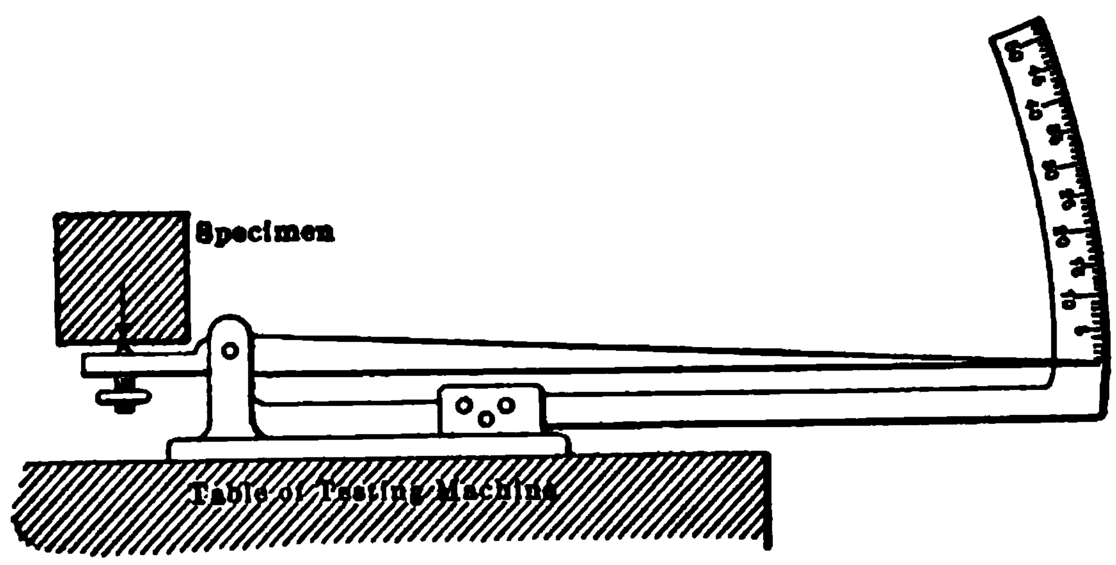


FIG. 374. — Deflectometer.

Resilience, often called the modulus of resilience, is a term used to represent the potential energy stored in a body; or, from another viewpoint, it is the amount of work that can be done by the body when relieved from a state of stress. More specifically, however, it is taken to mean in practice the **work, in foot-pounds,** done on a cubic inch of a material in stressing it to the **elastic limit.** For any value of the load the resilience is equal to the product of half the stress in pounds per square inch at the elastic limit times the distortion (usually the elongation) of the test-piece in feet per inch of length up to the elastic limit, the latter term being the space passed through. Values of resilience calculated as thus defined may be **checked** by comparing with the square of the unit stress at the elastic limit divided by 24 times the value of the modulus of elasticity (E).

Forms of Test-pieces for Tension Tests. Specimens for testing should be prepared with a great deal of care. Standard form for tests of **flat bars**[1] as well as for " coupons " cut from plates and structural shapes **in tension** is shown in **Fig. 375.** On such test-pieces marks one inch apart are usually made between the limits of the so-called gage length

[1] Adopted by American Society for Testing Material (1909).

which is generally 8 inches. A standard scale similar to the one in **Fig. 377** is of great assistance in marking a test-piece. At the left-hand end a percentage scale is shown from which the percentage of

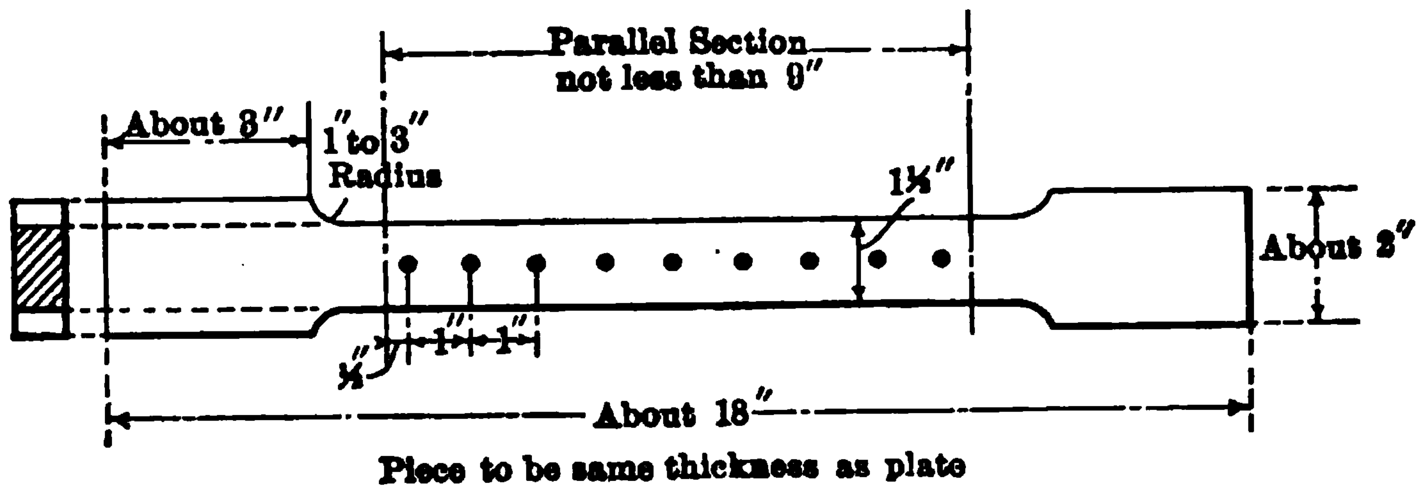


Fig. 375. — Standard Flat Bar for Tension Tests.

elongation, in a length of 8 inches, can be read directly. For testing round bars a shape shown in **Fig. 376** is sometimes used, making the middle portion ¾ inch in diameter. A special test piece of circular section only two inches long between the fillets where it is ½ inch in diameter

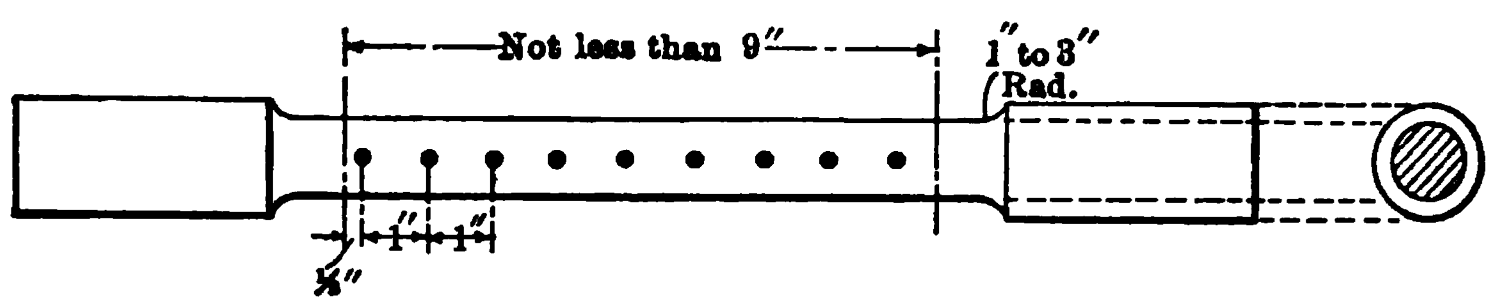


Fig. 376. — Round Bar sometimes used for Tension Tests.

is more generally recommended. The enlarged ends are ¾ inch in diameter and have screw threads turned on them.

Machine work on specimens for testing should be done carefully, so that the material is not torn or weakened in other ways. If there is

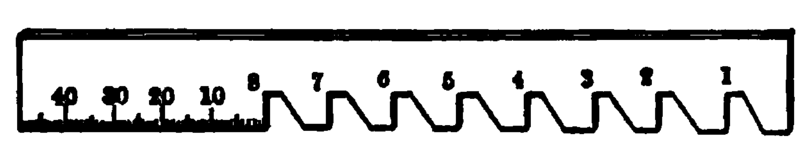


Fig. 377. — Scale for Marking Test-pieces.

any flaw, marked irregularity or other defect in the material, the test-piece should be rejected. After a test-piece has been "necked" and broken as shown in **Fig. 378**, the accurate measurement of the elongation is sometimes difficult. One method is to measure the elongation

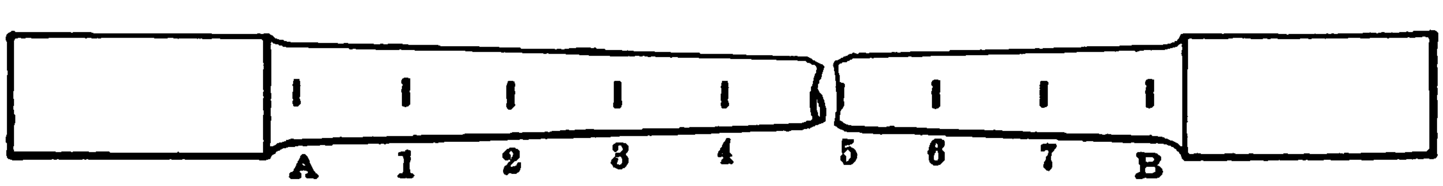


Fig. 378. — Test-piece after Rupture.

from the point of rupture toward each end. In materials in which the "necking" effect is very marked the measured amount of elongation will vary according to the distance of the fracture from the gage marks

A and B. If the fracture is midway between these marks then nearly all the elongation will be between these marks ; but if the fracture is near one of the gage marks then a great deal of the elongation will fall outside of the marks, so that the measured elongation is too small. To correct for these discrepancies mentioned, a so-called "equivalent elongation" may be calculated by the following method:

Assume that the standard test-piece, **Fig. 375**, has been divided originally into 8 equal spaces between the gage marks **A** and **B**, and that the nearest number of spaces between the points of fracture (**Fig. 378**) and the nearer gage mark is 3. The method may be stated that if the length between gage marks has been divided into **x** equal spaces and **y** is the **nearest** number of these spaces on the shorter portion between the point of rupture and the gage mark, then mark two points **M** and **N** on the **longer** portion, which are **y** and **1/2x** spaces, respectively, from the fracture. Place the two portions of the specimen as closely together as possible and measure from the gage mark in the short portion to the mark **M**. This distance, added to double the distance from **M** to **N**, gives the required total length after rupture. In this way the elongation of the "standard" length (**x** spaces) will be obtained, as if the fracture had occurred midway between the gage marks. To illustrate by the figure, there are 3 spaces on the **shorter** portion between the point of rupture and the gage mark **B**. The term **y** as defined above is therefore 3. Then the total length to be compared with the original is to be measured on the broken test-piece from 2 to **B**, corresponding to 6 spaces, plus twice the distance from 1 to 2, corresponding to the remaining 2 spaces to be accounted for.

Specifications are often made to require that the fracture shall be within the "middle third of the length."

Detailed Method for Tension Tests. A standard test-piece to be tested in **tension** should be without flaws or cracks, and furthermore the material should be monogeneous. Before putting it into the testing machine it should be carefully measured. With a scriber scratch the marks indicating one-inch divisions should be made with the "laying-off" gage. (**Fig. 377**.) Very light punch marks may then be made at each of the division marks accurately along the axis of the bar.[1] At each of these punch marks the diameter of the cross-section should be carefully measured with a micrometer caliper to thousandths of an inch. The outside punch marks, called the "gage marks," are often made a little heavier than the others, so that if an extensometer of the type illustrated

[1] Punch marks can be made accurately along the axis of a round bar by putting them at the "scratch" marks made with a scriber an inch apart along the length of the test-piece on the reflection of a "beam of light" on the bright surface of the test-piece.

in Fig. 372 is used, one of the thumbscrews supporting the clamps at each end can be set accurately but lightly into these marks. In this position the extensometer will measure accurately the elongation of the specimen between the gage marks. The testing machine to be used should then be balanced by adjusting the counterpoise provided for this purpose. It should be observed also that the "**check-nuts**" rest loosely on the table. As the load is increased, however, these nuts should be screwed down a little from time to time so that if the load is suddenly removed when the test-piece breaks, or should happen to slip out of the jaws or wedges holding it in place, the jar on the machine will be very much relieved.

After balancing the machine the test-piece should be placed carefully and vertically between the "jaws" or wedges in both the upper and lower heads, and the extensometer should be put in position if one is to be used. Start the machine at a low rate of speed until a load of about 1000 to 2000 pounds is indicated before taking any measurements of elongation. This first load is applied for the purpose of permitting the test-piece to assume a true central position, to allow for some slight slipping of the test-piece in the jaws before it becomes firmly gripped, and also to allow for possible irregularity in the adjustment of the extensometer or similar auxiliary apparatus. After this light load has been reached, during which time, doubtless for the kind of materials ordinarily tested, there will have been only a very small elongation, the load should be applied continuously and uniformly until the test-piece breaks, stopping only long enough, at the required intervals, to make the necessary observations of elongation and change of shape of the cross-section when "necking" begins. Increments of load are usually determined by taking one-tenth[1] the product obtained by multiplying the approximate estimated elastic limit of the material in pounds per square inch by the area of the cross-section in square inches.

Near the elastic limit it is often found desirable to take observations at intervals of 500 pounds up to the **yield point,** where suddenly the rate of elongation increases very rapidly. By taking these observations at very close intervals at the elastic limit, data are secured which make the curves to be plotted much more satisfactory.

In a **standard** test the stress, that is, the load, must never be **decreased** any appreciable amount when it is intended to apply still larger loads. A stress once applied must be maintained or increased continuously until the end of the test. Extensometers or other apparatus of delicate construction used for measuring the elongation should be removed

[1] In practice the increments of load in commercial tests are often one-half or one-third of the load at the elastic limit. For laboratory investigations it is not unusual to make the increments as small as one-twentieth of this load.

from the test-piece before the specimen is broken. It is customary to take off such apparatuses just after the elastic limit is reached, although, as a rule, they can be left in place relatively longer for materials that are ductile than for those that are hard and brittle. Micrometers used on extensometers provided with electrical connections are to be set for readings on one side at a time, advancing the screw device until the bell rings, indicating that the contact has been made. Then turn it back **just enough to stop the ringing of the bell,** and advance the screw on the other side until the bell rings again. After also turning back the micrometer screws **just enough to stop the ringing,** observations should be taken on both micrometers. Considerable time can be saved if, while these observations are being taken, the attendant having charge of the machine is meanwhile slowly advancing the load. The scale beam should be kept " floating " at all times during a test. Never run the balancing poise out on the scale beam beyond the point necessary to balance the beam. If the scale beam is carefully kept " floating " a point will be observed at from 50 to 75 per cent of the maximum load where the scale beam will fall, indicating **apparently** that it has been advanced too far. This point is called the **yield point.** It is defined as the stress at which the rate of elongation suddenly and rapidly increases.

Beyond the elastic limit, when the extensometer has been removed, the rate of elongation can be measured with considerable accuracy by means of a large machinist's dividers, with the points set accurately on the " **gage marks** " at the ends of the standard length.

The load when rupture occurs is not usually the **maximum.** When, therefore, considerable " necking " effect is observed, the poise on the scale beam should be watched closely and when the maximum load has been reached, indicated by the falling of the scale beam, this weight should be quickly observed and recorded and then the poise should be brought back to follow the decreasing load. It will be observed that **this part of the work is very interesting if it is done carefully.**

After the test-piece has been broken, stop the machine, remove the test-piece, clean and return the jaws or wedges to their proper places, **and leave the machine in good order.**

The broken ends of the test-piece should be joined carefully and the length between each of the marks which were originally one inch apart should be measured. Check the sum of these lengths with the over-all length between the gage marks. With a micrometer, or preferably a vernier caliper, measure as accurately as possible the diameter of the smallest area at the fracture. The fracture should be carefully examined to observe whether it is fibrous, granular or crystalline; whether coarse, fine or " silky;" whether cup-shaped, half-cup, or irregular in shape.

Curves and Calculations. Plot a curve with elongation per inch of length ("strain") as abscissas and stress in pounds per square inch as ordinates. This is the familiar "stress-strain" diagram.

Determine from the data obtained the modulus of elasticity (E),[1] the elastic limit, the maximum stress, the ultimate stress, per cent elongation in 8 inches, percentage elongation in 2 inches at the fracture, and percentage reduction in area at the fracture.

Plot a curve of elongation per inch, using for abscissas[2] the original length in inches and for ordinates the elongations measured for each inch between the gage marks.

REPORT OF TENSION TESTS

1. Date and names of observers..........
2. Material to be tested — specification..........
3. Makers and brand of material..........
4. Length between principal punch marks, before test..........
5. Length between principal punch marks, after test..........
6. Average width of test-piece, before test (6 readings)..........
7. Average width of test-piece, after test (6 readings)..........
8. Average thickness of test-piece, before test (9 readings)..........
9. Average thickness of test-piece after test (6 readings)..........

   Readings to be taken, for observations 10 and 11 for increments of 2000 lbs. on scale beam until an elongation of 0.40 in. is obtained between readings and then by increments of 1000 lbs. until elongation of 0.03 in. is obtained between readings, and finally by 500 lbs. until piece nears point of maximum strength, when readings should be taken as frequently as possible, keeping beam balanced all the time.

10. Total pull in pounds as shown by the machine..........
11. Corresponding total elongation in piece in inches..........
12. Yield point, pounds total (elastic limit)..........
13. Maximum load, pounds total..........
14. Breaking load, pounds total..........
15. Character of fracture, description and sketch..........

    The following results are to be computed:

16. Original cross-section, sq. in..........
17. Final cross-section, sq. in..........
18. Reduction in area in sq. in., and in per cent of original area..........
19. Final elongation, total and per cent..........
20. Stress (pounds per square inch of original area) at elastic limit, maximum load and breaking load.

[1] The modulus of elasticity can be determined also from the "stress-strain" diagram by calculating the value of the tangent for the angle between a line drawn through the origin parallel to the straight part of "stress-strain" curve, reading the scales of coördinates, of course, in the units of stress and elongation marked on the diagram. It is obvious that the value of the tangent of the angle in this case is the unit stress divided by the unit elongation, which is, by definition, the modulus of elasticity.

[2] If on the curve sheet the "inch marks" on the test-piece are indicated by equal divisions on the scale of abscissas, then the points showing elongation for each inch should be plotted midway between the division lines indicating the position of the "inch marks."

Tests in Compression. When a specimen of which the length is less than five times the smallest dimension is subjected to a load producing compression, it fails usually by **crushing.** Longer specimens fail usually by bending toward the side of least resistance. Two general classes of materials are frequently tested in compression: (1) **Brittle** materials, like brick, stone, wood, cement, cast iron, etc., which fail usually by **shearing,** and (2) **plastic** materials, like soft steel, wrought iron, copper, etc., which fail usually by a " flowing " of the metal. Because of the difficulty in measuring the deformations of short specimens of the plastic materials, and because the elastic limit in tension is invariably practically the same as in compression, these plastic materials are not often subjected to compressive loads. Methods to be explained here apply, therefore, particularly to materials like wood, brick, stone, and cast iron.

Detailed Method for Compression Tests of Short Test-pieces. Specimens of stone, cement, wood, or brick, of which the length is less than five times the smallest dimension, are **usually** provided in forms approximately cubes, although brick and wood are as often tested in the form of parallelopipeds similar to ordinary commercial bricks. Bearing surfaces of specimens of **stone, brick** and **cement** should be made as nearly flat and parallel as possible, and should then be covered with a thin layer of plaster of Paris. Sized paper in thin sheets should be placed on the bearing surfaces between the specimen and the plaster to prevent the absorption of water from the latter. In order to have the plaster set in a true surface the specimen is placed between the " heads " of the testing machine for about ten minutes after the movable cross-head (**B, Fig. 371**) has been lowered to press lightly on the plaster. For tests in compression the test-piece is placed on the table **T** and the load is applied by lowering the movable cross-head **B.**

Dimensions of test-pieces must be carefully measured and recorded before they are put into the testing machine; and, if any of them require the application of plaster of Paris, then the measurements must be made before the plaster is put on. After balancing the testing machine by means of the counterweight **with the test-piece on the table,** apply the load continuously until the specimen is fractured; or in the case of plastic materials until the deformation is quite noticeable. In general conduct the test in the same way as for tension,[1] except that the specimen is

[1] Measurements of the amount of compression (shortening) of the test-piece cannot be made directly, but must be made between points on the heads of the testing machine. If there is likely to be much yielding of the parts of the machine, the moving head should be lowered until its steel "compression plate" presses on the corresponding steel block on the table or lower platform with a force of about 1000 pounds. Now measure with micrometers the distance between the points on the two heads used for compression measurements, first with the load of 1000 pounds and then with additional increments of 1000 pounds up to considerably above the breaking load of the material

compressed instead of being stretched. If the material tested is cast iron or even hard stone or brick, precautions must be taken to protect persons near the machine from flying fragments. If the specimen begins to spall or flake off before it breaks down, the load corresponding as well as similar information should be recorded and placed in the tabulated report under " Remarks." Usually the specimen breaks down suddenly and the interior cone or pyramid in stone, brick, cement and cast iron will be plainly seen if the load has been " fairly " applied. In the case of tests of wood, this phenomenon will not be observed, but the lines of cleavage will usually show clearly a constant angle of shearing.

Detailed Method of Compression Tests of Long Pieces (Columns). When the length of a specimen to be tested in compression is greater than, at the most, ten times its least dimension, it fails invariably by bending toward the side of least resistance. The condition of the ends of such test-pieces should be as nearly as possible either fixed or perfectly free to turn. Either condition is, however, difficult to obtain. For test-pieces from 15 to 20 inches long usually an extensometer may be connected up to read the compression or shortening of the test-piece, if it is desired.[1] The observations will be taken in the same general way as for tension tests except that now the micrometer screws on the extensometer will approach each other, so that these screws must be turned back after taking a measurement by an amount greater than the compression that will be produced by the next increment of load.

Report on Compression Tests

1. Date and names of observers..
2. Kind of material..
3. Average thickness of test-piece (4 readings), inches........................
4. Average width of test-piece (4 readings), inches........................

The machine is to be started and kept running continuously until fracture takes place, the beam being kept balanced carefully all the time. Readings are to be taken, and calculations made therefrom as follows: In making these tests wood and brick will be used and two pieces of each kind are to be tested, with each kind of stress.

to be tested. From these data a correction curve should be plotted with which to correct the deflections observed when the specimen is tested.

When blocks of wood are to be tested in compression, readings of the micrometers should not be taken until a pressure of from 500 to 1000 pounds per square inch has been applied. This load will be required to crush the rough fibers.

[1] The lateral deflection along the neutral plane is sometimes determined by stretching a fine wire along the length of the specimen parallel to the neutral axis.

Kind of Stress	Tension.					
Kind of Wood	White Pine.		Yellow Pine.		Bricks.	
Number of Piece	1	2	1	2	1	2
Scale reading in pounds at time of fracture						
Character of fracture, sketch						
Cross-section from items 3 and 4, square inches						
Breaking stress lbs. per square inch for each piece						
Average breaking stress for each kind of wood						
Modulus of elasticity, lbs. per square inch						

Sketches, Curves and Calculations. Sketch the character of the fracture for each specimen tested, indicating, **for wood,** the direction of

FIG. 379. — Machine for Transverse Tests.

the grain. Previously, the original shape of the specimen should have been sketched and dimensioned.

Calculate the maximum unit stress.

If the material was suitable for the measurement of compression, plot "stress-strain" diagrams, and calculate the modulus of elasticity.

Transverse Bending Tests. The most common test by transverse or cross-bending is that of a beam usually of either wood or steel, of which the coefficient of elasticity and the "elastic curve" are desired. Deflections of such beams give the data needed. Such tests may be made with a testing machine like the one shown in **Fig. 371**, which is provided with supporting abutments marked in the figure **UU'**, and by inserting into the movable head the attachment for applying the load along a line across the beam rather than on a comparatively large area as in the tests already described when the load was applied directly by the flat surface of the cross-head **B.** Special transverse testing machines are, however, sometimes available. A machine of this kind is illustrated in **Fig. 379.**

In the Case of a **wooden** beam to be tested by loading at the middle, a fine steel wire should be stretched between two pins located as accurately as possible above the points of support and on the line of intersection of the neutral plane with the side of the beam. The wire should be fastened to one of these pins and allowed to hang over the other, being kept taut by means of a weight attached to the free end, **Fig. 380.**

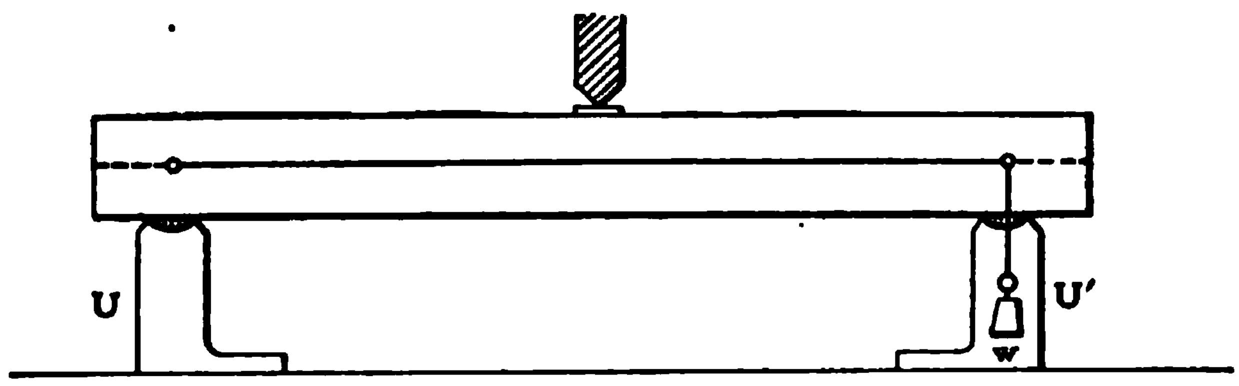


FIG. 380. — Device for Measuring the Deflection of a Wooden Beam.

A steel scale, preferably highly polished so that it will show the image of the wire, should be attached in any suitable way to the side of the beam, so that the edge along which the scale readings to be observed are marked will be exactly half-way between the two supports. The beam must be protected from indentation by the knife-edges by small bearing plates. The load should be applied centrally in increments to give approximately $\frac{1}{50}$ inch deflections to the elastic limit, and beyond to give deflections of approximately $\frac{1}{25}$ inch If it can be done successively, the deflections should be read without stopping the test; unless, of course, the permanent set is to be determined, when after each increment, the beam must be released from its load.

Curves and Calculations. Plot a curve taking the load applied in pounds for abscissas and deflections in inches for ordinates.

Sketch the character of the fracture.

Calculate the modulus of elasticity,[1] the modulus of rupture, and the stress in the outer fiber at the elastic limit from the curve.

Torsion Tests are made to determine the strength of a material to resist twisting forces. A typical machine for such tests is illustrated in **Fig. 381.** It consists in its essential parts of the frame **FF′**, the "jaw" heads **A** and **B** for gripping the rod **R** to be tested, and the system of weighing levers on which the poise **P** is balanced. The load is applied to the rod **R** by power through the gears shown connected to the head **B.**

Fig. 381. — Torsion Testing Machine.

The power is applied by means of the pulley and gears shown at the right-hand side, or may be applied by hand power by turning hand wheels. With hand power usually more satisfactory results can be obtained than with power applied mechanically, because the rate of twisting can be more closely regulated. The amount of twist or the

[1] The modulus of elasticity is calculated by the formula

$$E = \frac{w_e l^3}{48\, dI}. \quad \text{(141)}$$

The modulus of rupture from

$$f_u = \frac{w_u lc}{4\, I}, \quad \text{(142)}$$

and the stress in the outer fiber at the elastic limit by

$$f_e = \frac{w_e lc}{4\, I}, \quad \text{(143)}$$

when w_e = load at the elastic limit in pounds per square inch;
w_u = load at the point of rupture in pounds per square inch;
l = length of beam (span) in inches;
d = deflection in inches;
c = distance from the neutral axis to the outer fiber in inches;
I = moment of inertia, inch (4th power) units.

angular deformation is indicated by index-arms connected to opposite ends of the test-piece.

An autographic torsion testing machine operated by hand power by means of the crank is sometimes used. The movement of the crank tends to rotate the test-piece which at the opposite end of the machine is fastened to a pendulum carrying a heavy bob. The resistance of the pendulum and its weight measure the power applied, which is equal to the length of the lever arm times the sine of the angle of inclination times the constant weight of the bob.

Tests are made usually by increasing the twisting moment by increments of about 200 **inch-pounds**,[1] measuring for each increment the torsion angle.

Curves and Calculations. Plot a curve, using torsion angle for abscissas and twisting moment for ordinates. Calculate from this curve the unit stresses[2] (shearing) at the elastic limit, at the point of rupture, and also the maximum value of stress. Determine the torsion angle at the elastic limit and at the point of rupture, the helix angles,[3] and the modulus of elasticity **for torsion.**[4]

Impact Tests. Materials are tested by impact, usually by striking a test-piece with a weight allowed to fall upon it. Metals used in the manufacture of machinery and in railroad construction where it is likely to be subjected to shocks and blows are in many cases tested to determine the effect of the impact due to a blow.

Some testing machines for such tests are made like a pile-driver with

[1] If tests are made of large sections of high-grade material, like, for example, a shaft of nickel steel for a students' class, it is expensive to break many specimens, so that for this reason the twisting moment producing a maximum stress just inside the elastic limit is computed before making the test, and this value is not to be exceeded.

[2] The unit stress is calculated with the formula:

$$M/f_s = I_p/c, \quad \text{or} \quad f_s = Mc/I_p, \quad \ldots\ldots \quad (144)$$

where M is the torsional moment in inch-pounds, c is distance in inches from the neutral axis to the extreme fiber, and I_p is the polar moment of inertia. When $c = r$ (the radius) as in the case of a cylindrical test-piece, $I_p = \frac{1}{2}\pi r^4$.

[3] Torsion produces a peculiar arrangement of the outer fibers in the form of helices, as observed in broken test-pieces. Each one of these fibers makes an angle with its original position equal to its angular distortion α. Any particle on the surface is also moved through an angle β, having its vertex in the axis and in a plane perpendicular to the axis. Now if we neglect the functions of small angles, we can write approximately $l\alpha = r\beta$, where l is the effective length of the test-piece and r is the radius. The helix-angle $\alpha = r\beta/l$.

[4] The modulus of elasticity in torsion ("modulus of rigidity"),

$$E_s = f_s \div \alpha,$$

as above, then

$$E_s = \frac{f_s l}{r\beta}. \quad \ldots\ldots\ldots\ldots \quad (145)$$

the weight dropping vertically from a sort of gallows upon the test-piece. A more common form, however, of such machines consists of a pendulum provided with a heavy bob intended for delivering a blow on the middle of a test-piece in the shape of a bar, preferably of a rectangular section, held on two knife-edge supports attached to a heavy bedplate. Such machines are particularly designed for comparative tests of cast iron. They are provided with an arc concentric with the movement of the bob of the pendulum, graduated to read the vertical fall of the bob in feet. A tripping device is attached to the side of the graduated arc for permitting the bob to be supported and then dropped from any height within the limits of the machine. Since the deflection is very small, a device is usually supplied for magnifying it, and by means of a pencil-point traveling over a chart an autographic record is made of the deflections for each blow delivered by the bob. With such instruments the rebound of the test-piece and its permanent set must be carefully excluded from the measured deflection. One way to do this is to draw a " zero line " with the test-piece in place but before a blow is struck. Deflections and permanent set will then be measured on one side of this line and " rebounds " on the other.

To determine the center load to be applied that will be equivalent to the impact, the following symbols are used:

Let w_1 = weight of the bob in pounds;

h = the vertical distance it falls, in feet;

w_2 = the equivalent maximum center load, in pounds and

d = the deflection in feet, then

$$w_1 h = \tfrac{1}{2} w_2 d,$$

$$w_2 = \frac{2 w_1 h}{d} \quad \text{. . . . . (146)}$$

With this valve of w_2 the usual properties of the material may be calculated by formulas (141), (142) and (143), page 445.

Cement Tests. Cements are tested usually for (1) fineness; (2) time required for "setting"; (3) tensile strength; (4) specific gravity; (5) soundness or freedom from cracks after setting; (6) crushing strength; and (7) toughness or ability to resist blows. Tests for crushing strength (compression) are usually made by crushing cubical blocks in a testing machine designed for general tension and compression tests (see page 432). For tensile tests, however, special machines, designed particularly for testing cement, are generally used. Because of the nature of the material it is absolutely necessary that the power be applied in the line of the axis of the test-piece and also with steadiness and in increments as uniform as possible. There is a standard size and shape for test-pieces of cement, and they must be made in a certain prescribed way

in order that different tests may be compared. The standard briquette for testing (one square inch section) is shown in **Fig. 385**. The strength of the briquettes is affected by the time allowed for hardening, the amount of water used, and by the method of mixing the cement.

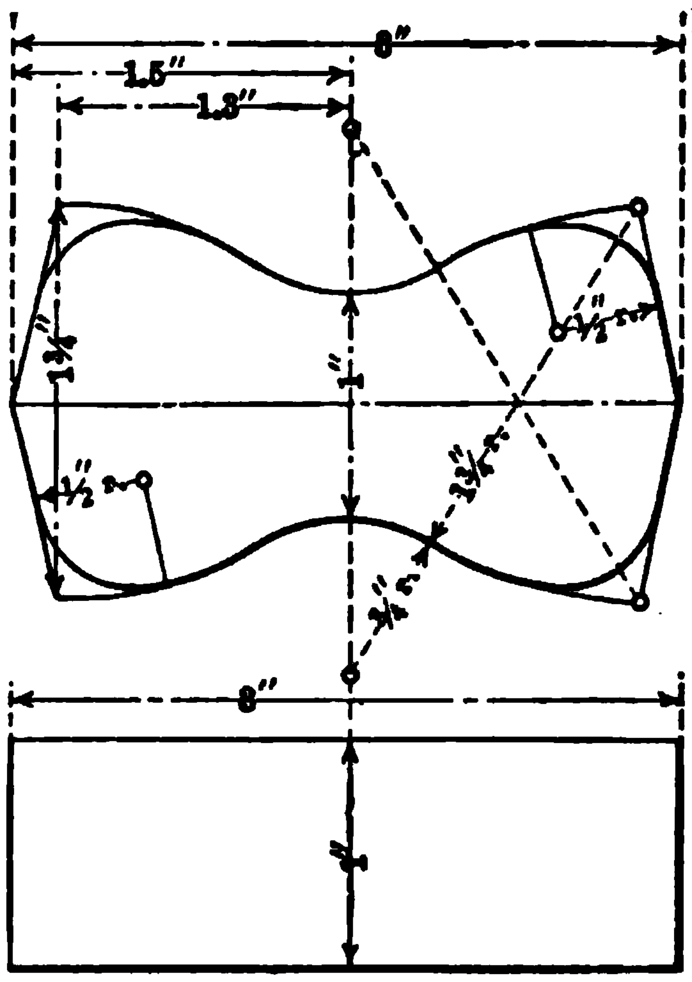


FIG. 385. — Standard Cement Briquette.

Power is applied in the **automatic** cement-testing machine in **Fig. 387**, by shot dropped from a cylindrical hopper into a pail supported on a scales. The briquette of cement being tested is held between two shackles or "holders" connected to a hand wheel used to regulate the distance between the shackles. When a briquette breaks the scale beam drops, and closes automatically a valve, stopping the delivery of shot into the pail. The operation of the machine may be described briefly as follows: Hang the hopper on the hook as shown and put enough shot into it to balance the counterpoise. Now put the briquette into the shackles and adjust the hand wheel so that the scale beam will rise nearly to the stop. When the valve is opened shot will begin to fall into the pail. The delivery of the shot into the pail should be slow. When the briquette has broken, the scale beam has dropped and the valve has been closed. The weight of shot collected in the pail shows the number of pounds required to break the briquette.

A non-automatic type of cement-testing machine is illustrated in **Fig. 388**. In this machine the power is applied by moving a hand wheel operating by means of a screw the system of levers transmitting the load to the briquette in the shackles. The tension produced by this load may obviously be balanced by weights applied to the scale beam above. In order to secure a very uniform and slow movement of the poise it is carried along the scale beam by a cord moved by a small crank. In applying the load the hand wheel should be moved as slowly and uniformly as possible to avoid a jerking motion. In the figure one of the standard briquette moulds, and a tray used for immersing the briquettes in water after they have set are shown. **Fig. 389** shows the proper position of the briquette in the supporting shackles.

Test of Cement for Fineness is made by determining the amount by weight of a given sample that will **not** pass through sieves with meshes of a standard size. The American Society of Civil Engineers recommends the use of sieves of 2500, 5476 and 10,000 meshes per square inch. Sieves

with approximately these meshings are known as Nos. 50, 80, and 100; that is, they have this number of meshes to the linear inch. A weighed sample of cement is first passed through a No. 50 sieve and the weight of that remaining in the sieve is recorded. That passing through is then put into the next sieve (No. 80) and the residue in this sieve is likewise weighed, while that passing through goes to the finest sieve (No. 100).

FIG. 387. — Automatic Cement Testing Machine.

Results of this test for fineness are expressed by the percentages that the various residues remaining in the sieves are of the original weight of the sample.

Unless cement is ground as fine as flour it has very little "binding power." The coarse particles are nearly as inert for "cementing" as sand.

Throughout America generally the following methods for testing cement adopted by the American Society of Civil Engineers in 1903 and 1904 and revised in January, 1909, are used:

Selection of Sample. The selection of the sample for testing is a detail that must be left to the discretion of the engineer; the number and the quantity to be taken from each package will depend largely on the importance of the work, the number of tests to be made and the facilities for making them.

Fig. 388. — Hand-operated Cement Testing Machine.

The sample shall be a fair average of the contents of the package; it is recommended that, where conditions permit, one barrel in every ten be sampled.

Samples should be passed through a sieve having twenty meshes per linear inch, in order to break up lumps and remove foreign material; this is also a very effective method for mixing them together in order to obtain an average.

Method of Sampling. Cement in barrels should be sampled through a hole made in the center of one of the staves, midway between the heads,

or in the head, by means of an auger or a sampling iron similar to that used by sugar inspectors. If in bags, it should be taken from surface to center.

Chemical Analysis. As a method to be followed for the analysis of cement, that proposed by the Committee on Uniformity in the Analysis of Materials for the Portland Cement Industry, of the New York Section of the Society for Chemical Industry, and published in *Engineering News*, Vol. 50, p. 60, 1903, is recommended.

Specific Gravity. The specific gravity of cement is lowered by adulteration and hydration; but the adulteration must be in considerable quantity to affect the results appreciably. Inasmuch as the differences in specific gravity are usually very small, great care must be exercised in making the determination.

The determination of specific gravity is most conveniently made with Le Chatelier's apparatus. This consists of a flask (*D*), **Fig. 390**, of 120 cu. cm. (7.32 cu. in.) capacity, the neck of which is about 20 cm.

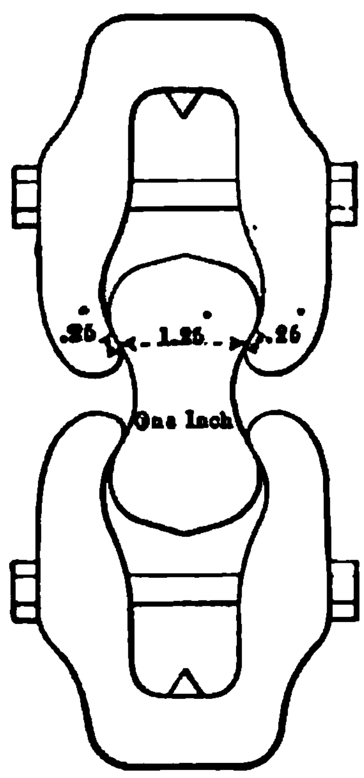


Fig. 389. — Briquette in Shackles.

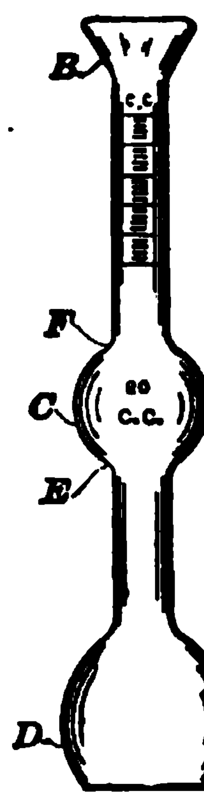


Fig. 390. — Le Chatelier's Specific Gravity Flask.

(7.87 in.) long; in the middle of this neck is a bulb (*C*), above and below which are two marks (*F*) and (*E*); the volume between these marks is 20 cu. cm. (1.22 cu. in.) The neck has a diameter of about 9 mm. (0.35 in.), and is graduated into tenths of cubic centimeters above the mark (*F*). Benzine (62° Baumé naphtha), or kerosene free from water, should be used in making the determination. Specific gravity can be determined in two ways:

(1) The flask is filled with either of these liquids to the lower mark (*E*), and 64 grams (2.25 oz.) of powder, cooled to the temperature of the liquid, is gradually introduced through the funnel (*B*) [the stem of which extends into the flask to the top of the bulb (*C*)], until the upper mark

(F) is reached. The difference in weight between the cement remaining and the original quantity (64 g.) is the weight which has displaced 20 cu. cm.

(2) The whole quantity of the powder is introduced, and the level of the liquid rises to some division of the graduated neck. This reading plus 20 cu. cm. is the volume displaced by 64 g. of the powder. The specific gravity is then obtained from the formula:

$$\text{Specific Gravity} = \frac{\text{Weight of Cement, in grams}}{\text{Displaced Volume, in cubic centimeters}}.$$

The flask during the operation is kept immersed in water in a jar (A), in order to avoid variations in the temperature of the liquid. The results should agree within 0.01. The determination of specific gravity should be made on the cement as received; and, should it fall below 3.10, a second determination should be made on the sample ignited at a low red heat.

A convenient method for cleaning the apparatus is as follows: The flask is inverted over a large vessel, preferably a glass jar, and shaken vertically until the liquid starts to flow freely; it is then held still in a vertical position until empty; the remaining traces of cement can be removed in a similar manner by pouring into the flask a small quantity of clean liquid and repeating the operation.

Fineness. It is generally accepted that the coarser particles in cement are practically inert, and it is only the extremely fine powder that possesses adhesive or cementing qualities. The more finely cement is pulverized, all other conditions being the same, the more sand it will carry and produce a mortar of a given strength. The degree of final pulverization which the cement receives at the place of manufacture is ascertained by measuring the residue retained on certain sieves. Those known as the No. 100 and No. 200 sieves are recommended for this purpose.

The sieves should be circular, about 20 cm. (7.87 in.) in diameter, 6 cm. (2.36 in.) high, and provided with a pan 5 cm. (1.97 in.) deep, and a cover. The wire cloth should be of brass wire having the following diameters: No. 100, 0.0045 in.; No. 200, 0.0024 in. This cloth should be mounted on the frames without distortion; the mesh should be regular in spacing and be within the following limits: No. 100, 96 to 100 meshes to the linear inch; No. 200, 188 to 200 meshes to the linear inch.

Fifty grams (1.76 oz.) or 100 grams (3.52 oz.) should be used for the test, and dried at a temperature of 100 deg. Cent. (212 deg. Fahr.) prior to sieving. The thoroughly dried and coarsely screened sample is weighed and placed on the No. 200 sieve, which, with pan and cover attached, is held in one hand in a slightly inclined position, and moved forward

and backward, at the same time striking the side gently with the palm of the other hand, at the rate of about 200 strokes per minute. The operation is continued until not more than one-tenth of 1 per cent passes through after one minute of continuous sieving. The residue is weighed, then placed on the No. 100 sieve and the operation repeated. The work may be expedited by placing in the sieve a small quantity of steel shot. The results should be reported to the nearest tenth of 1 per cent.

Normal Consistency. The use of a proper percentage of water in making the pastes[1] from which pats, tests of setting, and briquettes are made, is exceedingly important, and affects vitally the results obtained. The determination consists in measuring the amount of water required to reduce the cement to a given state of plasticity, or to what is usually designated as the normal consistency.

Method, Vicat Needle Apparatus. — This consists of a frame (*K*), Fig. 391, bearing a movable rod (*L*), with the cap (*A*) at one end, and at the other a cylinder 1 cm. (0.39 in.) in diameter, the cap, rod and cylinder weighing 300 grams (10.58 oz.). The rod, which can be held in any desired position by a screw (*F*) carries an indicator, which moves over a scale (graduated to centimeters) attached to the frame (*K*). The paste is held by a conical, hard-rubber ring (*I*), 7 cm. (2.76 in.) in diameter at the base, 4 cm. (1.57 in.) high, resting on a glass plate (*J*), about 10 cm. (3.94 in.) square.

FIG. 391. — Vicat Needle Apparatus.

In making the determination the same quantity of cement as will be subsequently used for each batch in making the briquettes, but not less than 500 grams, is kneaded into a paste, as described, and quickly formed into a ball with the hands, completing the operation by tossing it six times from one hand to the other, maintained 6 inches apart; the ball is then pressed into the rubber ring, through the larger opening, smoothed off, and placed (on its large end) on a glass plate and the smaller end smoothed off with a trowel; the paste, confined in the ring, resting on the plate, is placed under the rod bearing the cylinder, which is brought in contact with the surface and quickly released. The paste is of normal consistency when the cylinder penetrates to a point in the mass 10 mm. (0.39 in.)

[1] The term "paste" is used in this report to designate a mixture of cement and water, and the word "mortar" a mixture of cement, sand and water.

below the top of the ring. Great care must be taken to fill the ring exactly to the top. Trial pastes are made with varying percentages of water until the correct consistency is obtained. The Committee has recommended, as normal, a paste, the consistency of which is rather wet, because it believes that variations in the amount of compression to which the briquette is subjected in moulding are likely to be less with such a paste. Having determined in this manner the proper percentage of water required to produce a paste of normal consistency, the proper percentage required for the mortars is obtained from an empirical formula. The subject proves to be a very difficult one, and although the committee has given it much study, it is not yet prepared to make a definite recommendation. The Committee inserts the following table:

PERCENTAGE OF WATER FOR STANDARD MIXTURES

Neat	1 to 1	1 to 2	1 to 3	1 to 4	1 to 5	Neat	1 to 1	1 to 2	1 to 3	1 to 4	1 to 5
18	12.0	10.0	9.0	8.4	8.0	33	17.0	13.3	11.5	10.4	9.6
19	12.3	10.2	9.2	8.5	8.1	34	17.3	13.6	11.7	10.5	9.7
20	12.7	10.4	9.3	8.7	8.2	35	17.7	13.8	11.8	10.7	9.9
21	13.0	10.7	9.5	8.8	8.3	36	18.0	14.0	12.0	10.8	10.0
22	13.3	10.9	9.7	8.9	8.4	37	18.3	14.2	12.2	10.9	10.1
23	13.7	11.1	9.8	9.1	8.5	38	18.7	14.4	12.3	11.1	10.2
24	14.0	11.3	10.0	9.2	8.6	39	19.0	14.7	12.5	11.2	10.3
25	14.3	11.6	10.2	9.3	8.8	40	19.3	14.9	12.7	11.3	10.4
26	14.7	11.8	10.3	9.5	8.9	41	19.7	15.1	12.8	11.5	10.5
27	15.0	12.0	10.5	9.6	9.0	42	20.0	15.3	13.0	11.6	10.6
28	15.3	12.2	10.7	9.7	9.1	43	20.3	15.6	13.2	11.7	10.7
29	15.7	12.5	10.8	9.9	9.2	44	20.7	15.8	13.3	11.9	10.8
30	16.0	12.7	11.0	10.0	9.3	45	21.0	16.0	13.5	12.0	11.0
31	16.3	12.9	11.2	10.1	9.4	46	21.3	16.1	13.7	12.1	11.1
32	16.7	13.1	11.3	10.3	9.5						

	1 to 1	1 to 2	1 to 3	1 to 4	1 to 5
Cement...........................	500	333	250	200	167
Sand.............................	500	666	750	800	833

Time of Setting. The object of this test is to determine the time which elapses from the moment water is added until the paste ceases to be fluid and plastic (called the "initial set"), and also the time required for it to acquire a certain degree of hardness (called the "final" or "hard set"). The former of these is the more important since, with the commencement of setting, the process of crystallization or hardening is said to begin. As a disturbance of this process may produce a loss of strength, it is desirable to complete the operation of mixing and moulding or incorporating the mortar into the work before the cement begins to set. It is usual to measure arbitrarily the beginning and end

of the setting by the penetration of weighted wires of given diameters. For this purpose the Vicat Needle, **Fig. 391**, should be used. In making the test, a paste of normal consistency is moulded and placed under the rod (*L*), bearing the cap (*A*) at one end and the needle (*H*), 1 mm. (0.039 in.) in diameter, at the other, weighing 300 grams (10.58 oz.). The needle is then carefully brought in contact with the surface of the paste and quickly released.

The setting is said to have commenced when the needle ceases to pass a point 5 mm. (0.20 in.) above the upper surface of the glass plate, and is said to have terminated the moment the needle does not sink visibly into the mass. Test-pieces should be stored in moist air during the test; this is accomplished by placing them on a rack over water contained in a pan and covered with a damp cloth, the cloth to be kept away from them by means of a wire screen; or they may be stored in a moist box or closet. Care should be taken to keep the needle clean, as the collection of cement on the sides of the needle retards the penetration, while cement on the point reduces the area and tends to increase the penetration. The determination of the time of setting is only approximate, being materially affected by the temperature of the mixing water, the temperature and humidity of the air during the test, the percentage of water used, and the amount of molding the paste receives.

Standard Sand. The Committee recognizes the grave objections to the standard quartz now generally used, especially on account of its high percentage of voids, the difficulty of compacting in the moulds, and its lack of uniformity; it has spent much time in investigating the various natural sands which appeared to be available and suitable for use. For the present, the Committee recommends the natural sand from Ottawa, Ill., screened to pass a sieve having 20 meshes per linear inch and retained on a sieve having 30 meshes per linear inch; the wires to have diameters of 0.0165 and 0.0112 in., respectively, *i.e.*, half the width of the opening in each case. Sand having passed the No. 20 sieve shall be considered standard when not more than 1 per cent passes a No. 30 sieve after one minute's continuous sifting of a 500 g. sample.[1]

Form of Briquette. While the form of the briquette recommended by a former Committee of the Society is not wholly satisfactory, this Committee is not prepared to suggest any change, other than rounding off the corners by curves of ½-in. radius, **Fig. 385**.

Molds. The molds should be made of brass, bronze, or some equally non-corrodible material, having sufficient metal in the sides to prevent

[1] The Sandusky Portland Cement Company, of Sandusky, Ohio, has agreed to undertake the preparation of this sand, and to furnish it at a price only sufficient to cover the actual cost of preparation.

spreading during molding. Gang molds, which permit molding a number of briquettes at one time, are preferred by many to single molds, since the greater quantity of mortar that can be mixed tends to produce greater uniformity in the results. The type shown in Fig. 393 is recommended. The molds should be wiped with an oily cloth before using.

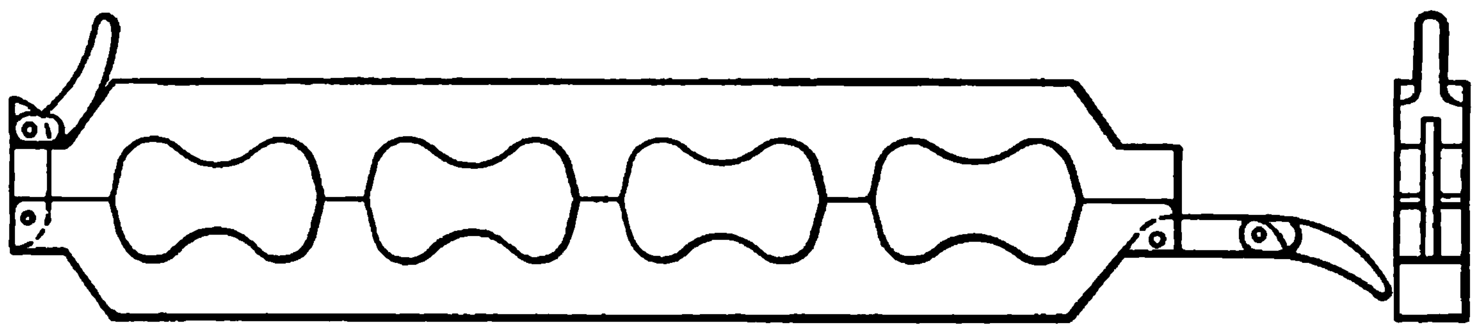


FIG. 393.— Briquette Mold.

Mixing. All proportions should be stated by weight; the quantity of water to be used should be stated as a percentage of the dry material. The metric system is recommended because of the convenient relation of the gramme and the cubic centimeter. The temperature of the room and the mixing water should be as near 21 deg. Cent. (70 deg. Fahr.) as it is practicable to maintain it. The sand and cement should be thoroughly mixed dry. The mixing should be done on some non-absorbing surface, preferably plate glass. If the mixing must be done on an absorbing surface it should be thoroughly dampened prior to use. The quantity of material to be mixed at one time depends on the number of test pieces to be made; about 1000 grams (35.28 oz.) makes a convenient quantity to mix, especially by hand methods. The material is weighed and placed on the mixing table, and a crater formed in the center, into which the proper percentage of clean water is poured; the material on the outer edge is turned into the crater by the aid of a trowel. As soon as the water has been absorbed, which should not require more than one minute, the operation is completed by vigorously kneading with the hands for an additional one and a half minutes, the process being similar to that used in kneading dough. During the operation of mixing, the hands should be protected by gloves, preferably of rubber.

Molding. Having worked the paste or the mortar consistency it is at once placed in the molds by hand. The molds should be filled immediately after the mixing is completed, the material pressed in firmly with the fingers and smoothed off with a trowel without mechanical ramming; the material should be heaped up on the upper surface of the mold, and, in smoothing off, the trowel should be drawn over the mold in such a manner as to exert a moderate pressure on the excess material. The mold should be turned over and the operation repeated.

A check upon the uniformity of the mixing and molding is afforded by weighing the briquettes just prior to immersion, or upon removal

from the moist closet. Briquettes which vary in weight more than 3 per cent from the average should not be tested.

Storage of Test-pieces. During the first twenty-four hours after molding, the test-pieces should be kept in moist air to prevent them from drying out. A moist closet or chamber is so easily devised that the use of the damp cloth should be abandoned if possible. Covering the test-pieces with a damp cloth is objectionable, as commonly used, because the cloth may dry out unequally, and, in consequence, the test-pieces are not all maintained under the same condition. Where a moist closet is not available, a cloth may be used and kept uniformly wet by immersing the ends in water. It should be kept from direct contact with the test-pieces by means of a wire screen or some similar arrangement. A moist closet consists of a soapstone or slate box, or a metal-lined wooden box—the lining being covered with felt and this felt kept wet. The bottom of the box is so constructed as to hold water, and the sides are provided with cleats for holding glass shelves on which to place the briquettes. Care should be taken to keep the air in the closet uniformly moist. After twenty-four hours in moist air, the test-pieces for longer periods of time should be immersed in water maintained as near 21 deg. Cent. (70 deg. Fahr.) as practicable; they may be stored in tanks or pans, which should be of non-corrodible material.

Tensile Strength. The tests may be made on any standard machine. A solid metal clip, as shown in **Fig. 389** (page 451), is recommended. This clip is to be used without cushioning at the points of contact with the test specimen. The bearing at each point of contact should be ¼ in. wide, and the distance between the center of contact on the same clip should be 1¼ in. Test pieces should be broken as soon as they are removed from the water. Care should be observed in centering the briquettes in the testing machine, as cross-strains, produced by improper centering, tend to lower the breaking strength. The load should not be applied too suddenly, as it may produce vibration, the shock from which often breaks the briquette before the ultimate strength is reached. Care must be taken that the clips and the sides of the briquette be clean and free from grains of sand or dirt, which would prevent a good bearing. The load should be applied at the rate of 600 lbs. per min. The average of the briquettes of each sample tested should be taken as the test, excluding any results which are manifestly faulty.

Constancy of Volume. The object is to develop those qualities which tend to destroy the strength and durability of a cement. As it is highly essential to determine such qualities at once, tests of this character are for the most part made in a very short time, and are known, therefore, as accelerated tests. Failure is revealed by cracking, checking, swelling, or disintegration, or all of these phenomena. A cement which remains

perfectly sound is said to be of constant volume. Tests for constancy of volume are divided into two classes: (1) normal tests, or those made in either air or water maintained at about 21 deg. Cent. (70 deg. Fahr.), and (2) accelerated tests, or those made in air, steam, or water at a temperature of 45 deg. Cent. (115 deg. Fahr.) and upward. The test-pieces should be allowed to remain 24 hours in moist air before immersion in water or steam, or preservation in air. For these tests pats, about 7½ cm. (2.95 in.) in diameter, 1¼ cm. (0.49 in.) thick at the center, and tapering to a thin edge, should be made upon a clean glass plate [about 10 cm. (3.94 in.) square], from cement paste of normal consistency.

Normal Test. A pat is immersed in water maintained as near 21 deg. Cent. (70 deg. Fahr.) as possible for 28 days, and observed at intervals. A similar pat, after 24 hours in moist air, is maintained in air at ordinary temperature and observed at intervals.

Data regarding tests of neat cement and mortar briquettes may be tabulated in a form similar to the following:

FORM FOR CEMENT TESTS

1. Date, and names of observers.
2. Name and kind of cement.
3. Makers and location of plant.
4. Distinguishing mark on briquettes.
5. Date of mixing and time.
6. Temperature of room at time of mixing, deg. F.
7. Temperature of water at time of mixing, deg. F.
8. Conditions of setting: as to time briquettes were left in dampened air before immersión, etc.
9. Activity of the cement or time of initial and final setting.[1]
10. Fineness of grinding.

	Kind of Briquette.	Neat.									Sand.								
12 13 14	Composition of Briquettes.	Per Cent of Cement .... Per Cent of Water									Per Cent of Cement.............. Per Cent of Water................. Per Cent of Sand.................								
	No. of Briquettes.	7 Day.				28 Day.				Remarks	7 Day.				28 Day.				Remarks
		1	2	3	4	5	6	7	8		1	2	3	4	5	6	7	8	
15	Time of test.																		
16	Breaking strength, lbs.																		
17	Appearance of fracture —give sketch of each here.																		
18	Temp. of room at time of test. deg. F.																		

In reporting the results of these experiments it is important that the effect of different percentages of water sand, etc., and time of immersion, be fully discussed.

[1] See pages 454 and 455.

Accelerated Test. A pat is exposed in any convenient way in an atmosphere of steam, above boiling water, in a loosely closed vessel,[1] for 5 hours. To pass these tests satisfactorily, the pats should remain firm and hard, and show no signs of cracking, distortion or disintegration. Should the pat leave the plate, distortion may be detected best with a straight-edge applied to the surface which was in contact with the plate.

[1] The apparatus recommended for this test is shown with a dimensioned drawing in *Proc. A.S.C.E.*, vol. 35, No. 2 (1909).

CHAPTER XXIV

OUTLINES OF SUGGESTED TESTS

1. Calibration and Adjustment of Pressure Gage. Reference, pages 7–22.

Apparatus. Dead weight gage tester, standard weights, and gage to be tested.

Method. Take readings at intervals of 5 lbs. per sq. in., up and down. Spin platform and weights to eliminate friction. Remove the indicating needle with special jack and take off dial. Sketch parts in interior of gage. Reset needle to read correctly in part of scale most used.[1] Attach needle firmly. Repeat readings up and down for new calibration.

Report. Explain methods of adjustment. Tabulate as on page 21.

Curves.[2] (See page 22.) Draw curves only for final condition of gage.

2. Calibration and Adjustment of Vacuum Gage. Reference, pages 22–24 and 236.

Apparatus. Mercury U-tube, aspirator (ejector) or air pump, and gage to be tested.

Method. Take readings at intervals of 2 in. vacuum, up and down. If there is water or other impurity on mercury column correction must be made. (Reset needle if so instructed.)

Report. Explain details of method used. Tabulate as on page 21, omitting second column and writing " ins. vac. " for "pressure, lbs. per sq. in."

Curves. Similar to instructions for test No. 1.

3. Thermometer Calibration for Range Above 212° F. Reference, pages 29–39.

Apparatus. Steam gage used in test No. 1, thermometers (a standard high-reading thermometer is sometimes used to check the results), barometer and steam tables (pages 468–470).

Method. Read thermometer being tested (and " standard " if one is used) at intervals of approximately 5 lbs. per sq. in. on the gage calibrated in test No. 1. Be sure the steam is not superheated and allow at least 5 min. after final adjustments of valves before readings are taken.

Report. Sketch with a simple line drawing the interior of apparatus used with necessary piping connections. Tabulate as on page 37. Calculate " stem " corrections when necessary.

Curves. (See page 36.) " True " temperatures are from steam tables.

4. Use and Calibration of a Planimeter. Reference, pages 74–87, 141.

Apparatus. Polar planimeter, large compass, scale, and micrometer.

Method. 1. Measure length of tracing-arm from pivot to tracing-point and diameter of rolling or graduated wheel to check accuracy for reading areas in sq. in.

2. Calculate length of tracing-arm so that one revolution of graduated wheel will indicate 8 sq. in.

3. Find area of zero circle by at least two methods (see pages 77, 80).

4. Determine average error in per cent of instrument by first measuring and then calculating the area of circles of 1, 2, and 3 in. diameter (see page 87, footnote).

[1] If other instructions are not given assume two-thirds maximum graduation of dial.

[2] Arrangement of coördinates for curves is throughout to make them most applicable for use; that is, in the use of a curve the given quantity should be read on the scale of abscissas and the value to be found will then be obtained from the scale of ordinates.

Measure each area three times and take average. Mark percentage error + if instrument reads too small and − if too large.

5. Determine indicated horse power for an indicator card. Scale of indicator spring =40 lbs. per sq. in. (i.e., No. 40 spring); area of piston = 66 sq in.; length of stroke = 1 ft.; and r.p.m. = 200.

Report. Record data, method of measurements, and calculations. Tabulate results.

5. Calibration of Indicator Springs in Compression. Reference, pages 92–106, 112–120, 136–144. Read Precautions for Care of Indicator, pages 103–106.

Apparatus. Indicator, set of springs, and indicator spring tester.

Method. 1. Test perpendicularity of motion of pencil-point to atmospheric line. (Footnote, page 105.)

2. Obtain at least two good calibration cards for each spring similar to Fig. 121, for increasing and decreasing pressures. Obviously unless the cards obtained for a spring are alike, the work is not successful. Take increments of 5 lbs. per sq. in. for all springs up to and including the "40 lb." spring. For springs of higher scale take increments of 10 lbs. per sq. in. Lines of maximum pressure on calibration card should be about 1¾ ins. above the atmospheric line.

When a plunger type of tester is used, that is, similar in principle to Fig. 118, the diameter of the plunger must be measured with a micrometer and the relation accurately calculated between the weight and the unit pressure applied to the indicator in lbs. per sq. in.

Examine at least two types of indicators. Insert the spring and study the adjustment of the height of the pencil.

When work with an indicator is finished, always remove the "piston" spring and thoroughly clean all parts, inside and outside.

Report. Tabulate and draw curves as directed on page 119. Calculate the true scale of spring from the average of four equidistant points on this curve. Explain calculations. Discuss any discrepancies in data.

If a "40-lb." spring has been calibrated, assume this was used in obtaining the indicator card used in test No. 4, and calculate the corrected indicated horse power.

5a. Calibration of Indicator Springs in Tension (for use with Vacuum). Reference, pages 118, 119.

Apparatus. Indicator, springs, and special tester.

Method and report same as for test No. 5.

6. Study of Reducing Motions. Reference, pages 121–138.

Apparatus. Steel scale and drawing instruments.

Method and Report. Examine reducing motions in laboratory. Design an accurate device for an engine as designated by instructor.

7. Calorimeter Tests, Use and Comparison. Reference, pages 55–73.

Apparatus. Separating calorimeter, glass beaker (or graduate), pail with cover having a hole for insertion of hose for condensing steam, platform scales, watch, throttling calorimeter, thermometers, steam pressure gage, barometer and monkey wrench.

Method. Calibrate steam gage. Connect the separating and throttling calorimeters by means of standard sampling nipples (see page 56) to the same vertical steam pipe where they will both take steam of the same quality. Allow steam to blow through both calorimeters until the temperatures in the throttling type have reached a maximum value and the other has become thoroughly heated and enough water has collected to bring the level in the water gage glass up to, or a little above, the zero on the scale. Make the condensing pail about two-thirds full of water. Obtain this weight of water by weighings. When all conditions are satisfactory, put the hose through which steam has been discharging from the separating calorimeter into the hole in the cover of the pail, and at the same time observe reading of scale of water gage. Read temperatures

and pressures every three minutes. Run the test until an appreciable volume of steam discharges from the hole in the cover of the pail. Then throw steam tube out of pail and read the level in the water gage. Again weigh pail and contents. Calibrate the scale of the water gage by removing and weighing water from the separating calorimeter between any two levels within the limits of the scale.

Make one test at each of four steam pressures above 60 lbs. per sq. in. gage as designated by instructor.

Calculate the quality of steam roughly by the charts on pages 59 or 61 during the progress of the tests and immediately check with data from separating calorimeter.

Report. Tabulate all observed and calculated data. Sketch and describe fully the calorimeters used. State in detail all operations in performing test. Discuss relative accuracy of results.

8. Test of Platform Scales.

Apparatus. Platform scales to be tested, 12 in. steel scale, graduated to $\frac{1}{100}$ in. and standard 50- or 100-lb. weights.

Method. Platform scales are probably used more in engineering work about a power plant than any other measuring device, and usually young engineers do not very well understand their operation. They consist essentially of a device by which a load is applied to a system of levers, of long and short arms, arranged so that a load on the platform can be balanced by weights applied at the end of a final lever called the beam, or by shifting a poise along the length of this latter lever. Essentially it is like the weighing device shown in Fig. 370. This weighing beam is usually placed on an upright post at one end of the platform.

1. Take off the platform and measure the length of all the lever arms between knife-edges to the nearest $\frac{1}{100}$ in. Draw simple line sketch showing all arms and lengths.

2. *Observe the means provided to adjust the beam to read zero.*

3. Observe sensitiveness of scales by finding range through which poise can be shifted without appreciably disturbing the balance.

4. Calibrate the scales after assembling by placing standard weights on the platform and observing the reading on the beam when balanced. After calibration shift scales around roughly and observe result by a recalibration. Test also by placing loads in middle of platform as a scales should be used, and then at any of the sides.

Report. 1. Plot results of the calibration with observed weights as abscissas and standard weights as ordinates.

2. Discuss effect of rough handling on the calibration, and whether an accurate calibration of a scales made before shipping can be considered absolutely reliable later. Explain effect of non-central loading on the platform.

3. From the measured lever arms calculate the weight of poise for two different positions on the beam.

4. Calculate the weight of a poise for an additional beam which would indicate readings to $\frac{1}{10}$ the smallest division on the beam of this scale. (Note that the additional weight of the extra beam could be balanced by a larger adjusting counterweight.)

9. Oil Tests: Viscosity, Flash and Burning Points, and Specific Gravity. Reference, pages 395–404.

Apparatus. Viscosimeter, flash tester, hydrometer, 2 thermometers, 2 glass graduates, Bunsen burner, matches, wax tapers, test-tube, and watch.

Method. 1. Determine flash and burning points of oil in flash tester, with top closed for flash point and open for burning point. To check, use at least two samples of the same kind of oil. Never use a second time a sample that has been heated. Why? Determine the chill point of the oil if ice is available.

2. With the orifice of viscosimeter and inner cup thoroughly cleaned determine the time required for 50 cu. cm. of water at "room" temperature to flow through this

orifice, starting from the level of the tip of the hook-gage. Make 3 tests and take average. Determine similarly the time required for 50 cu. cm. of the oil at temperatures of 80, 110, 140 and 170° F. to flow through the same orifice when starting at the same level.

3. Determine the specific gravity of the oil in both the Baumé and the "specific gravity" scales at about 80°, 100°, 120°, and 140° F. (Avoid pouring hot oils into cold glass vessels as they are likely to be broken.)

Report. Tabulate results of the various tests. Plot (1) temperatures as abscissas and viscosities as ordinates, (2) temperatures as abscissas and specific gravity as ordinates, both curves having the same abscissas on the same sheet.

Discuss suitableness or unsuitableness of this oil for various services.

10. **Oil Tests** (Cont'd): **Coefficient of Friction and Temperature Rise of Bearings.** Reference, pages 404–405.

Apparatus. Pendulum oil-tester, thermometer, wooden strut with knife-edge at end, sensitive platform scales, spirit level, and steel scale graduated to $\frac{1}{100}$ in.

Method. Support the pendulum in a horizontal position (as determined by a spirit level) on a strut, provided with a knife-edge at the bottom where it rests on a platform scales. Determine the weight of the pendulum alone in this position. Measure also the effective length of the lever arm from the center of the journal to a vertical line through the knife-edge. Repeat weighing and measurement of length of arm for several points along the pendulum. Remove the pendulum and determine its weight accurately. Measure the length (l) and diameter (d) of the journal so as to calculate the projected area 2ld of the bearing surface. Compute the constant $\frac{W'R}{r}$, and by setting the pendulum at various angles determine whether the scale is correctly graduated. In order to get satisfactory results the bearing and shaft must be perfectly clean and smooth. Calibrate spring in pendulum.

Operate the machine at constant speed with pressures of 50, 100, 150 and 200 lbs. per sq. in. on the bearing. Record the arc of deflection, temperatures of bearing and of room, speed, and rate of oil feed (usually 3 drops per minute).

Report. Plot curves of bearing pressure (lbs. per sq. in.) as abscissas and coefficients of friction and temperatures as ordinates.

Calculate velocity of rubbing faces in ft. per min. In machines arranged like ordinary shop bearings where all the load is on the bottom half of the bearing this velocity is calculated on the basis of only half the circumference.

Discuss possible errors in the data as shown by the curves.

11. **Proximate Analysis of Coal.** Reference, pages 228–234.

Apparatus. Crucibles, Bunsen burners, matches, coal crusher, mortar and pestle, 20-mesh sieve, 2 air-tight bottles, drying oven, air-blast lamp (or an equivalent), rubber tubing, watch, desiccator, and "chemical" balance sensitive to 1 part in 1000 of amount weighed.

Method. That of Am. Chem. Soc. (pages 229–231) *or* of A.S.M.E. (pages 231–233) as directed by instructor. Inquire about location of mine from which coal was taken. Make duplicate determinations to check values of moisture and volatile matter.

Report. Record all data. Determine percentages of moisture, volatile matter, fixed carbon and ash in coal "as received." Also percentages of volatile matter, fixed carbon and ash in dry coal. Discuss results by comparing with analyses of coals from same district as given in mechanical engineers hand-books, etc.

12. **Calorific Value of Coal.** Reference, pages 210–222.

A. Apparatus. Bomb calorimeter with platinum crucible, mortar and pestle, 100-mesh sieve, oxygen tank with pipe connections to fit threads on bomb, accurate

calorimeter, thermometer, pail,[1] scales for weighing water, fine iron wire for ignition, "chemical" balance sensitive to $\frac{1}{1000}$ gram, briquetting machine,[2] monkey wrench and calorimeter spanner.

B. Apparatus. Same as "A" except Parr calorimeter is used instead of bomb type, and absolutely pure sodium peroxide is used instead of oxygen gas.

Method. See pages 210–215 for "A" and pages 217–219 for "B." Make at least two determinations.

Report for "A" or "B." Describe apparatus used and procedure in detail. Calculate heating value of coal tested in B.t.u. per lb. "as received," also heating value per lb. dry coal and per lb. combustible in same units. Record all data. Discuss results by comparing with heating value of coals from same district as given in mechanical engineer's hand-books, etc.

13. **Calorific Value of Gas.** Reference, pages 222–227.

Apparatus. Junkers' calorimeter (with gas burner), 2 calorimeter thermometers, 2 ordinary thermometers, 2 glass graduates, large pail, platform scales, "wet" gas meter, gas regulator, glass U-tube for gas pressure, barometer, and rubber tubing.

Method. See pages 223–226. Read thermometers every two minutes.

Make at least two determinations which should check within 1 per cent.

Report. Tabulate all observations. Explain calculations.

Determine "higher" and "lower" heating values of the gas per cu. ft. (1) at conditions of test and (2) at standard conditions (see pages 223).

Sketch apparatus used. Discuss possible errors in method.

14. **Calorific Value of Oil.** Reference, pages 222–227.

Apparatus. Junkers' calorimeter (with oil lamp and chemical balance for its attachment) and hydrometer. Otherwise same as for test No. 13.

Method. See pages 226–227. Collect water from calorimeter during time required for burning exactly 50 grams of oil. Determine specific gravity of oil used. Make at least two determinations.

Report. Tabulate all observations. Calculate "higher" and "lower" heating values per lb. of oil. Discuss results by comparing with data given in books on gas and oil engines or in mechanical engineer's hand-books.

15. **Analysis of Flue Gas.** Reference, pages 235–252, 281–283.

Apparatus for sampling and chemical absorption of gases as directed by instructor.

Method. See pages 235–245. Make at least two analyses from the same sample of gas which should check throughout within $\frac{1}{10}$ per cent.

Report. Tabulate data and results (calculated as percentages).

From the average results of the analysis calculate number of lbs. air required to burn (1) a lb. of carbon and (2) a lb. of dry coal, assuming the dry coal contains 83 per cent carbon, 3 per cent hydrogen, 4 per cent oxygen and 10 per cent earthy matter.

16. **Study of Brakes.** Reference, pages 147–163.

Apparatus. Steel scale, calipers, and drawing instruments.

Method and Report. Design prony brake of type stated by instructor for absorbing b.h.p. at. . .r.p.m. Show calculations for diameter at root of thread of tightening bolt, and for determining capacity of scales required to take the pressure of the brake. Calculate also weight of water required per hour for cooling the brake, assuming 10 per cent of heat dissipated by radiation.

Answer the following questions:

1. If water were shut off from the brake what damage would result?

[1] Instead of the pail and scales a large glass graduate is often used to measure the water in cu. cm.

[2] Briquetting the coal is not required by A.S.M.E. recommendations.

2. Why is it a very bad practice to stop the engine with the full load on the brake?
3. Why should a safety-cord be attached to the arm of the brake?
4. What is the effect of putting oil on the rim of the brake pulley? Of putting on water?

If there is any doubt as to the proper answers, run an engine with a Prony brake attached to find out by experience.

17. Calibration of a Transmission Dynamometer. Reference, pages 164–174.

Apparatus. Dynamometer and its weights steel scale, hand speed counter and watch.

Method. Measure length of lever arms. Observe condition of dash-pot as explained on page 168. Remove the brake from its pulley and make a series of runs; that is, without load at various speeds and observe the corresponding readings of instrument. Attach the brake to its pulley and make a series of tests at three different speeds with net loads on the brake for each speed of approximately (1)...(2)...(3)..., and (4) ...lbs. Determine " zero " load or tare of brake, lbs., and length of brake arm, ft.

For each test record: (1) Gross brake load, lbs., (2) net brake load, lbs., (3) reading of dynamometer (theoretically, lbs.), (4) r.p.m., and (5) names of observers.

Report. Examine construction of dynamometer and sketch arrangement of lever arms (and gears, if any). For each speed calculate ft.-lbs. per minute corresponding to readings of dynamometer and plot curve with these as abscissas and ft.-lbs. by brake as ordinates. Plot also curve for each speed of reading of dynamometer as abscissas and net brake load, lbs. as ordinates.

(For Flather's and Morin's dynamometers plot height of autographic diagram, ins. as abscissas with ft.-lbs. by brake as ordinates.)

18. Belting Tests. Reference, pages 392–394.

Apparatus. Belt tester, steel tape, 2 hand speed counters, watch, calibrated scales for weighing, and 3 ft. scale for lengths.

Method. See pages 392–394. Determine " zero " load or tare on each scale. Adjust the distance between driving and driven pulleys when at rest so that there is initial tension in belt of 25 lbs. per in. of width. Run tests at a given constant speed of the driver at varying loads until the belt begins to slip. Make about five runs for this value of initial belt tension. Each test should be of at least 5 min. duration during which time both speed counters should be read continuously. Observe for each test also average brake load, lbs. and tension reading, lbs.

Make similar sets of runs with initial belt tensions of 50 and 75 lbs. per in. of width.

Measure diameter of either driving or the driven pulleys (inches) and also arc of contact of belt on this pulley (inches). Calculate ratio of arc in inches to radius in inches. Measure length, width and thickness of belt.

Report. Calculate slip of belt, coefficient of friction, b.h.p., and efficiency of transmission.

For each value of initial tension plot b.h.p. as abscissas and as ordinates (1) slip (per cent) and (2) coefficient of friction. Also initial tension of belt as abscissas and b.h.p. as ordinates.

Discuss results in their application to shop practice.

19. Test of Hoists. Reference, pages 391–392.

Apparatus. Hoists to be tested, spring balance, scale with graduations marked very plainly.

Method. Same pages as reference.

Report. Sketch hoists used. On the same curve sheet plot for each hoist a curve of load lifted (lbs.) as abscissas and efficiency as ordinates.

20. Mechanical Efficiency Test of Steam Engine. Determinations of Indicated and Brake Horse Power. Reference, pages 136–142, 147, 150, 284.

Apparatus. Steam engine indicators, steam pressure gage, prony or rope brake, hand speed counter, watch, scale about 3 ft. long, platform scales and planimeter, cans of cylinder and engine oils.

Method. Put spring in indicator, oil its piston with cylinder oil and the joints of pencil motion with porpoise or similar light oil. Adjust indicator parts so that there is no lost motion in pencil mechanism. Attach firmly to engine cylinder. Adjust cord so that it will have normal tension when engine is turned over both dead centers. Fill engine cylinder lubricator with cylinder oil and all oil cups with engine oil. Adjust feed of all oiling devices. Measure effective brake arm and "zero" load or tare of brake (see pages 148–151).

Vary net load on brake by increments of 50 lbs. up to maximum engine will carry without slowing down. Run each test for 12 min., taking all indicator cards and readings of r.p.m. and gross brake reading, lbs. every 3 min. Measure as many indicator cards as possible while test is in progress and compare at least a few values of indicated with brake h.p. before test is finished. This is done to check the accuracy of the work. **Always clean the engine and shut off the lubricator and oil cups when finishing a test.**

Calculate engine and brake constants (see pages 143 and 148).

Report. Tabulate data and calculated results as on page 284. Examine and sketch the reducing motion. Explain whether or not it gives an accurate reduction.

Plot with i.h.p. as abscissas the following as ordinates: (1) b.h.p., (2) mech. effic. (per cent), (3) r.p.m. and (4) friction h. p. If friction horse power has not constant values discuss reasons.

21. Setting of Plain D-slide Valve on Steam Engine. Reference, pages 285–288, 289.

Apparatus. Steel scale, monkey wrench, trammels or large machinist's dividers, chalk, drawing-board and instruments and indicators.

Method. See pages 285–287. Remove steam chest cover and valve from its stem. Measure face of valve and ports. Make dimensioned drawing like Fig. 297 (page 285) and Zeuner valve diagram.[1] Adjust laps and eccentric as explained in reference. Replace steam-chest cover. Attach indicators and take diagrams. Compare with ideal diagram obtained from the Zeuner drawing.

If indicator diagrams are not satisfactory make the adjustments needed.

Report. Explain in detail procedure. Discuss each step in making adjustments.

22. Setting of Corliss Valve on Steam Engine. Reference, pages 288–293.

Apparatus. Monkey wrenches, steel scale, plumb bob and line and indicators.

Method. Adjust (1) wrist plate, (2) all reach-rods for given laps, (3) rocker. Put engine on dead center and set eccentric for a given lead. Readjust reach-rods. Take indicator cards and readjust valves.

23. Volumetric Clearance of Engine. Reference, pages 293–294.

Apparatus. Pails with cocks near the bottom, rubber tubing, funnels, small platform scales for weighing water in pails, trammels, chalk, monkey wrench, watch, wooden blocks to cover parts of engine and rubber packing.

Method. Remove steam-chest cover and cover ports with blocks on top of rubber packing. Set engine on dead-center (see page 286, footnote). Continue procedure as in reference above.

24. Boiler Test. Reference, pages 258–283.

Apparatus. Steam gage, draft gage, barometer, watch, wrenches, steam calorimeter (with manometer if needed), thermometers with maximum graduation below 240° F. for (1) external air, (2) boiler room, (3) feed-water entering boiler[2] and (4) make-up water; above 240° F. for (1) temperature of steam at steam nozzle (discharge from boiler) and (2) steam calorimeter, platform scales for (1) coal, (2) ashes and (3) feed-

[1] See treatises on Steam Engines. Zeuner diagrams are most useful for valve setting and Bilgram diagrams and best for designing.

[2] For plants operating with an economizer a thermometer for higher temperatures would be required.

water, large tanks for feed-water, standard weights, thermo-couple for flue temperature, flue gas apparatus for sampling and analyzing, jars or cans for samples of coal and ash, means for marking level of water in gage glass, large closed cans for accumulating samples of coal and ashes.

Method. See reference. Calibrate gage and thermometers. Plot a graphical log as test proceeds (see page 268). Run boiler leakage test for 3 hrs. at normal boiler pressure (see page 339) before regular boiler test begins.

Report. Tabulate all data. Use A.S.M.E. "short" *or* "long" form as directed by instructor.

25. **For Economy Tests of Steam Engines, Steam Turbines, Complete Steam Power Plants, Gas Engines, Oil Engines, Gas Producers, as well as Pump Tests, Injector Tests, Air Compressor Tests, Air Lift Tests, Ventilating Fan Test, and Refrigerating Plant Tests.** See detailed instructions and A. S. M. E. codes, pages 294–314, 317–328, (329–335), 336–340, 345–363,372–376, 386–388, 409–424, 428–430.

26. **Tests of the Strength of Materials of Construction.** See Chapter XXIII for detailed methods and reports.

APPENDIX

Two sets of tables which the author has found useful for "rough and ready" calculations are given on the following pages. Table I is a short table of the more important properties of saturated steam. This table has been taken with permission from Allen and Bursley's *Heat Engines.*

Table II gives for various common substances the specific gravity, density (weight per cubic foot), specific heat and coefficients of expansion per degree Fahrenheit, both linear and volumetric. Coefficient of volumetric expansion is three times the linear coefficient of expansion.

TABLE I

PROPERTIES OF SATURATED STEAM

ENGLISH UNITS

Abs. Pressure, Pounds per Sq. In.	Temperature, Degrees F.	Heat of the Liquid.	Latent Heat of Evaporation.	Total Heat of Steam.	Specific Volume, Cu. Ft. per Pound.	Density, Pounds per Cu. Ft.	Abs. Pressure. Pounds per Sq. In.
p	t	q or h	r or L	H	v	$\frac{1}{v}$	p
.0886	32	0	1072.6	1072.6	3301.0	.000303	.0886
.2562	60	28.1	1057.4	1085.5	1207.5	.000828	.2562
.5056	80	48.1	1046.6	1094.7	635.4	.001573	.5056
1	101.8	69.8	1034.6	1104.4	333.00	.00300	1
2	126.1	94.1	1021.4	1115.5	173.30	.00577	2
3	141.5	109.5	1012.3	1121.8	118.50	.00845	3
4	153.0	120.9	1005.6	1126.5	90.50	.01106	4
5	162.3	130.2	1000.2	1130.4	73.33	.01364	5
6	170.1	138.0	995.7	1133.7	61.89	.01616	6
7	176.8	144.8	991.6	1136.4	53.58	.01867	7
8	182.9	150.8	988.0	1138.8	47.27	.02115	8

PROPERTIES OF SATURATED STEAM—*Continued*

ENGLISH UNITS

Abs. Pressure, Pounds per Sq. In.	Temperature, Degrees F.	Heat of the Liquid.	Latent Heat of Evaporation.	Total Heat of Steam.	Specific Volume, Cu. Ft. per Pound.	Density, Pounds per Cu. Ft.	Abs. Pressure, Pounds per Sq. In.
p	t	q or h	r or L	H	v	$\frac{1}{v}$	p
9	188.3	156.3	984.8	1141.1	42.36	.02361	9
10	193.2	161.2	981.7	1142.9	38.38	.02606	10
11	197.7	165.8	978.9	1144.7	35.10	.02849	11
12	202.0	170.0	976.3	1146.3	32.38	.03089	12
13	205.9	173.9	973.9	1147.8	30.04	.03329	13
14	209.6	177.6	971.6	1149.2	28.02	.03568	14
14.7	212.0	180.1	970.0	1150.1	26.79	.03733	14.7
15	213.0	181.1	969.4	1150.5	26.27	.03806	15
16	216.3	184.5	967.3	1151.8	24.77	.04042	16
17	219.4	187.7	965.3	1153.0	23.38	.04277	17
18	222.4	190.6	963.4	1154.0	22.16	.04512	18
19	225.2	193.5	961.5	1155.0	21.07	.04746	19
20	228.0	196.2	959.7	1155.9	20.08	.04980	20
21	230.6	198.9	958.0	1156.9	19.18	.05213	21
22	233.1	201.4	956.4	1157.8	18.37	.05445	22
23	235.5	203.9	954.8	1158.7	17.62	.05676	23
24	237.8	206.2	953.2	1159.4	16.93	.05907	24
25	240.1	208.5	951.7	1160.2	16.30	.0614	25
26	242.2	210.7	950.3	1161.0	15.71	.0636	26
27	244.4	212.8	948.9	1161.7	15.18	.0659	27
28	246.4	214.9	947.5	1162.4	14.67	.0682	28
29	248.4	217.0	946.1	1163.1	14.19	.0705	29
30	250.3	218.9	944.8	1163.7	13.74	.0728	30
31	252.2	220.8	943.5	1164.3	13.32	.0751	31
32	254.1	222.7	942.2	1164.9	12.93	.0773	32
33	255.8	224.5	941.0	1165.5	12.57	.0795	33
34	257.6	226.3	939.8	1166.1	12.22	.0818	34
35	259.3	228.0	938.6	1166.6	11.89	.0841	35
36	261.0	229.7	937.4	1167.1	11.58	.0863	36
37	262.6	231.4	936.3	1167.7	11.29	.0886	37
38	264.2	233.0	935.2	1168.2	11.01	.0908	38
39	265.8	234.6	934.1	1168.7	10.74	.0931	39
40	267.3	236.2	933.0	1169.2	10.49	.0953	40
41	268.7	237.7	931.9	1169.6	10.25	.0976	41
42	270.2	239.2	930.9	1170.1	10.02	.0998	42
43	271.7	240.6	929.9	1170.5	9.80	.1020	43
44	273.1	242.1	928.9	1171.0	9.59	.1043	44
45	274.5	243.5	927.9	1171.4	9.39	.1065	45
46	275.8	244.9	926.9	1171.8	9.20	.1087	46
47	277.2	246.2	926.0	1172.2	9.02	.1109	47
48	278.5	247.6	925.0	1172.6	8.84	.1131	48
49	279.8	248.9	924.1	1173.0	8.67	.1153	49
50	281.0	250.2	923.2	1173.4	8.51	.1175	50
51	282.3	251.5	922.3	1173.8	8.35	.1197	51
52	283.5	252.8	921.4	1174.2	8.20	.1219	52
53	284.7	254.0	920.5	1174.5	8.05	.1241	53
54	285.9	255.2	919.6	1174.8	7.91	.1263	54
55	287.1	256.4	918.7	1175.1	7.78	.1285	55
56	288.2	257.6	917.9	1175.5	7.65	.1307	56
57	289.4	258.8	917.1	1175.9	7.52	.1329	57
58	290.5	259.9	916.2	1176.1	7.40	.1351	58
59	291.6	261.1	915.4	1176.5	7.28	.1373	59
60	292.7	262.2	914.6	1176.8	7.17	.1394	60
61	293.8	263.3	913.8	1177.1	7.06	.1416	61

PROPERTIES OF SATURATED STEAM — *Continued*

ENGLISH UNITS

Abs. Pressure, Pounds per Sq. In.	Temperature, Degrees F.	Heat of the Liquid.	Latent Heat of Evaporation.	Total Heat of Steam.	Specific Volume, Cu. Ft. per Pound.	Density, Pounds per Cu. Ft.	Abs. Pressure, Pounds per Sq. In.
p	t	q or h	r or L	H	v	$\frac{1}{v}$	p
62	294.9	264.4	913.0	1177.4	6.95	.1438	62
63	295.9	265.5	912.2	1177.7	6.85	.1460	63
64	297.0	266.5	911.5	1178.0	6.75	.1482	64
65	298.0	267.6	910.7	1178.3	6.65	.1503	65
66	299.0	268.6	910.0	1178.6	6.56	.1525	66
67	300.0	269.7	909.2	1178.9	6.47	.1547	67
68	301.0	270.7	908.4	1179.1	6.38	.1569	68
69	302.0	271.7	907.7	1179.4	6.29	.1591	69
70	302.9	272.7	906.9	1179.6	6.20	.1612	70
71	303.9	273.7	906.2	1179.9	6.12	.1634	71
72	304.8	274.6	905.5	1180.1	6.04	.1656	72
73	305.8	275.6	904.8	1180.4	5.96	.1678	73
74	306.7	276.6	904.1	1180.7	5.89	.1699	74
75	307.6	277.5	903.4	1180.9	5.81	.1721	75
76	308.5	278.5	902.7	1181.2	5.74	.1743	76
77	309.4	279.4	902.1	1181.5	5.67	.1764	77
78	310.3	280.3	901.4	1181.7	5.60	.1786	78
79	311.2	281.2	900.7	1181.9	5.54	.1808	79
80	312.0	282.1	900.1	1182.2	5.47	.1829	80
81	312.9	283.0	899.4	1182.4	5.41	.1851	81
82	313.8	283.8	898.8	1182.6	5.34	.1873	82
83	314.6	284.7	898.1	1182.8	5.28	.1894	83
84	315.4	285.6	897.5	1183.1	5.22	.1915	84
85	316.3	286.4	896.9	1183.3	5.16	.1937	85
86	317.1	287.3	896.2	1183.5	5.10	.1959	86
87	317.9	288.1	895.6	1183.7	5.05	.1980	87
88	318.7	288.9	895.0	1183.9	5.00	.2002	88
89	319.5	289.8	894.3	1184.1	4.94	.2024	89
90	320.3	290.6	893.7	1184.3	4.89	.2045	90
91	321.1	291.4	893.1	1184.5	4.84	.2066	91
92	321.8	292.2	892.5	1184.7	4.79	.2088	92
93	322.6	293.0	891.9	1184.9	4.74	.2110	93
94	323.4	293.8	891.3	1185.1	4.69	.2131	94
95	324.1	294.5	890.7	1185.2	4.65	.2152	95
96	324.9	295.3	890.1	1185.4	4.60	.2173	96
97	325.6	296.1	889.5	1185.6	4.56	.2194	97
98	326.4	296.8	889.0	1185.8	4.51	.2215	98
99	327.1	297.6	888.4	1186.0	4.47	.2237	99
100	327.8	298.4	887.8	1186.2	4.430	.2257	100
101	328.6	299.1	887.2	1186.3	4.389	.2278	101
102	329.3	299.8	886.7	1186.5	4.349	.2299	102
103	330.0	300.6	886.1	1186.7	4.309	.2321	103
104	330.7	301.3	885.6	1186.9	4.270	.2342	104
105	331.4	302.0	885.0	1187.0	4.231	.2364	105
106	332.0	302.7	884.5	1187.2	4.193	.2385	106
107	332.7	303.4	883.9	1187.3	4.156	.2407	107
108	333.4	304.1	883.4	1187.5	4.119	.2428	108
109	334.1	304.8	882.8	1187.6	4.082	.2450	109
110	334.8	305.5	882.3	1187.8	4.047	.2472	110
111	335.4	306.2	881.8	1188.0	4.012	.2493	111
112	336.1	306.9	881.2	1188.1	3.977	.2514	112
113	336.8	307.6	880.7	1188.3	3.944	.2535	113
114	337.4	308.3	880.2	1188.5	3.911	.2557	114
114.7	337.9	308.8	879.8	1188.6	3.888	.2572	114.7

PROPERTIES OF SATURATED STEAM — *Continued*

ENGLISH UNITS

Abs. Pressure, Pounds per Sq. In.	Temperature, Degrees F.	Heat of the Liquid.	Latent Heat of Evaporation.	Total Heat of Steam.	Specific Volume, Cu. Ft. per Pound.	Density, Pounds per Cu. Ft.	Abs. Pressure, Pounds per Sq. In.
p	t	q or h	r or L	H	v	$\frac{1}{v}$	p
115	338.1	309.0	879.7	1188.7	3.878	.2578	115
116	338.7	309.6	879.2	1188.8	3.846	.2600	116
117	339.4	310.3	878.7	1189.0	3.815	.2621	117
118	340.0	311.0	878.2	1189.2	3.748	.2642	118
119	340.6	311.7	877.6	1189.3	3.754	.2663	119
120	341.3	312.3	877.1	1189.4	3.725	.2684	120
121	341.9	313.0	876.6	1189.6	3.696	.2706	121
122	342.5	313.6	876.1	1189.7	3.667	.2727	122
123	343.2	314.3	875.6	1189.9	3.638	.2749	123
124	343.8	314.9	875.1	1190.0	3.610	.2770	124
125	344.4	315.5	874.6	1190.1	3.582	.2792	125
126	345.0	316.2	874.1	1190.3	3.555	.2813	126
127	345.6	316.8	873.7	1190.5	3.529	.2834	127
128	346.2	317.4	873.2	1190.6	3.503	.2855	128
129	346.8	318.0	872.7	1190.7	3.477	.2876	129
130	347.4	318.6	872.2	1190.8	3.452	.2897	130
131	348.0	319.3	871.7	1191.0	3.427	.2918	131
132	348.5	319.9	871.2	1191.1	3.402	.2939	132
133	349.1	320.5	870.8	1191.3	3.378	.2960	133
134	349.7	321.0	870.4	1191.4	3.354	.2981	134
135	350.3	321.6	869.9	1191.5	3.331	.3002	135
136	350.8	322.2	869.4	1191.6	3.308	.3023	136
137	351.4	322.8	868.9	1191.7	3.285	.3044	137
138	352.0	323.4	868.4	1191.8	3.263	.3065	138
139	352.5	324.0	868.0	1192.0	3.241	.3086	139
140	353.1	324.5	867.6	1192.1	3.219	.3107	140
141	353.6	325.1	867.1	1192.2	3.198	.3128	141
142	354.2	325.7	866.6	1192.3	3.176	.3149	142
143	354.7	326.3	866.2	1192.5	3.155	.3170	143
144	355.3	326.8	865.8	1192.6	3.134	.3191	144
145	355.8	327.4	865.3	1192.7	3.113	.3212	145
146	356.3	327.9	864.9	1192.8	3.093	.3233	146
147	356.9	328.5	864.4	1192.9	3.073	.3254	147
148	357.4	329.0	864.0	1193.0	3.053	.3275	148
149	357.9	329.6	863.5	1193.1	3.033	.3297	149
150	358.5	330.1	863.1	1193.2	3.013	.3319	150
152	359.5	331.2	862.3	1193.5	2.975	.3361	152
154	360.5	332.3	861.4	1193.7	2.939	.3403	154
156	361.6	333.4	860.5	1193.9	2.903	.3445	156
158	362.6	334.4	859.7	1194.1	2.868	.3487	158
160	363.6	335.5	858.8	1194.3	2.834	.3529	160
162	364.6	336.6	858.0	1194.6	2.801	.3570	162
164	365.6	337.6	857.2	1194.8	2.768	.3613	164
166	366.5	338.6	856.4	1195.0	2.736	.3655	166
168	367.5	339.6	855.5	1195.1	2.705	.3697	168
170	368.5	340.6	854.7	1195.3	2.674	.3739	170
172	369.4	341.6	853.9	1195.5	2.644	.3782	172
174	370.4	342.5	853.1	1195.6	2.615	.3824	174
176	371.3	343.5	852.3	1195.8	2.587	.3865	176
178	372.2	344.5	851.5	1196.0	2.560	.3907	178
180	373.1	345.4	850.8	1196.2	2.532	.3949	180
182	374.0	346.4	850.0	1196.4	2.506	.3990	182
184	374.9	347.4	849.3	1196.7	2.480	.4032	184
186	375.8	348.3	348.5	1196.8	2.455	.4074	186

PROPERTIES OF SATURATED STEAM — *Concluded*

ENGLISH UNITS

Abs. Pressure, Pounds per Sq. In.	Temperature, Degrees F.	Heat of the Liquid.	Latent Heat of Evaporation.	Total Heat of Steam.	Specific Volume, Cu. Ft. per Pound.	Density Pounds per Cu. Ft.	Abs. Pressure, Pounds per Sq. In.
p	t	q or h	r or L	H	v	$\frac{1}{v}$	p
188	376.7	349.2	847.7	1196.9	2.430	.4115	188
190	377.6	350.1	847.0	1197.1	2.406	.4157	190
192	378.5	351.0	846.2	1197.2	2.381	.4200	192
194	379.3	351.9	845.5	1197.4	2.358	.4242	194
196	380.2	352.8	844.8	1197.6	2.335	.4284	196
198	381.0	353.7	844.0	1197.7	2.312	.4326	198
200	381.9	354.6	843.3	1197.9	2.289	.4370	200
202	382.7	355.5	842.6	1198.1	2.268	.4411	202
204	383.5	356.4	841.9	1198.3	2.246	.4452	204
206	384.4	357.2	841.2	1198.4	2.226	.4493	206
208	385.2	358.1	840.5	1198.6	2.206	.4534	208
210	386.0	358.9	839.8	1198.7	2.186	.4575	210
212	386.8	359.8	839.1	1198.9	2.166	.4618	212
214	387.6	360.6	838.4	1199.0	2.147	.4660	214
216	388.4	361.4	837.7	1199.1	2.127	.4700	216
218	389.1	362.3	837.0	1199.3	2.108	.4744	218
220	389.9	363.1	836.4	1199.5	2.090	.4787	220
222	390.7	363.9	835.7	1199.6	2.072	.4829	222
224	391.5	364.7	835.0	1199.7	2.054	.4870	224
226	392.2	365.5	834.3	1199.8	2.037	.4910	226
228	393.0	366.3	833.7	1200.0	2.020	.4950	228
230	393.8	367.1	833.0	1200.1	2.003	.4992	230
232	394.5	367.9	832.3	1200.2	1.987	.503	232
234	395.2	368.6	831.7	1200.3	1.970	.507	234
236	396.0	369.4	831.0	1200.4	1.954	.511	236
238	396.7	370.2	830.4	1200.6	1.938	.516	238
240	397.4	371.0	829.8	1200.8	1.923	.520	240
242	398.2	371.7	829.2	1200.9	1.907	.524	242
244	398.9	372.5	828.5	1201.0	1.892	.528	244
246	399.6	373.3	827.8	1201.1	1.877	.532	246
248	400.3	374.0	827.2	1201.2	1.862	.537	248
250	401.1	374.7	826.6	1201.3	1.848	.541	250
275	409.6	383.7	819.0	1202.7	1.684	.594	275
300	417.5	392.0	811.8	1203.8	1.547	.647	300
350	431.9	407.4	798.5	1205.9	1.330	.750	350

TABLE II

Properties of Common Substances

	Specific Gravity.	Weight per Cubic Foot, Lbs.	Weight of One Cubic Inch, Lb.	Specific Heat.	Coefficient of Expansion per Deg. F.	
					Linear.	Volumetric.
Aluminum	2.60	161	.095	.212	.000011	.000033
Bismuth	9.82	613	.353	.031	.000008	.000024
Boxwood, along or across grain	.....	.....	.....	.....	.000002	.000006
Brass	8.10	503	.293	.094	.00001	.00003
Cement	2.24	140	.083	.20	.000008	.000024
Copper	8.79	545	.318	.097	.000009	.000028
Coal (anthracite)	1.43	88.7	.058	.241	.....	.....
Coke	1.00	62.4	.037	.203	.....	.....
Gasoline	.68	42.4	.....	.....	.....	.....
Glass	2.89	180.7	.105	.198	.000005	.000014
Gold	19.26	1200	.697	.032	.000008	.000024
Ice (at 32° F.)	.92	57.5	.033	.504	.....	.....
Iron (cast)	7.5	465	.271	.130	.000006	.000018
Iron (wrought)	7.74	582	.280	.110	.000007	.000021
Lead	11.35	708	.411	.031	.000016	.000048
Limestone	3.16	197	.114	.217	.....	.....
Mercury (at 32° F.)	13.60	849	.492	.033	.000033	.000100
Nickel	8.90	547	.321	.109	.000007	.000020
Pine (white), along grain	.55	34	.020	.65	.0000025	.000008
Pine (white), across grain	.....	.....	.....	.....	.000020	.000080
Platinum	21.5	1342	.779	.032	.000005	.000015
Porcelain	.....	.....	.....	.....	.000002	.000006
Oak, along grain	.....	.....	.....	.....	.000003	.000009
Silver	10.47	653	.379	.056	.000011	.000033
Steel	7.83	486	.292	.116	.000007	.000020
Tin	7.29	452	.264	.056	.000012	.000035
Zinc	7.19	445	.260	.095	.000016	.000048

TABLE III

Metric Conversion Table

Millimeters × .03937 = inches.
Millimeters ÷ 25.4 = inches.
Centimeters × .3937 = inches.
Centimeters ÷ 2.54 = inches.
Meters × 39.37 = inches. (Act of Congress.)
Meters × 3.281 = feet.
Meters × 1.094 = yards.
Kilometers × .6214 = miles.
Kilometers ÷ 1.6093 = miles.
Kilometers × 3280.8 = feet.
Square Millimeters × .00155 = sq. inches.
Square Millimeters ÷ 645.2 = sq. inches.
Square Centimeters × .155 = sq. inches.
Square Centimeters ÷ 6.452 = sq. inches.
Square Meters × 10.764 = sq. feet.
Square Kilometers × 247.1 = acres.
Hectare × 2.471 = acres.
Cubic Centimeters ÷ 16.387 = cubic inches.
Cubic Meters × 35.314 = cubic feet.

Cubic Meters × 1.308 = cubic yards.
Cubic Meters × 264.2 = gallons (231 cu. in.)
Liters × 61.023 = cubic in. (Act of Congress.)
Liters × .2642 = gallons (231 cu. in.)
Liters ÷ 3.785 = gallons (231 cu. in.)
Liters ÷ 28.317 = cubic feet.
Hectoliters × 3.531 = cubic feet.
Hectoliters × 2.838 = bushels (2150.42 cu. in.)
Hectoliters × .1308 = cubic yards.
Hectoliters × 26.42 = gallons (231 cu. in.)
Grams × 15.432 = grains (Act of Congress.)
Grams × 981. = dynes.
Grams ÷ 28.35 = ounces avoirdupois.
Grains ÷ 15.432 = grams.
Grains ÷ 7000 = pounds.
Joule × .7373 = foot pounds.
Kilograms × 2.2046 = pounds.
Kilograms × 35.27 = ounces avoirdupois.
Kilograms ÷ 907.2 = tons (2,000 lbs.)
Kilogr. per sq. cent. × 14.223 = lbs. per sq. in.
Kilogrammeters × 7.233 = foot lbs.
Kilo per sq. Meter × .672 = lbs. per sq. foot.
Kilo per Cu. Meter × .0624 = lbs. per cu. ft.
Kilowatts × 1.34 = Horse Power.
Watts ÷ 746. = Horse Power.
Watts × .7373 = foot pounds per second.
Calorie × 3.968 = B.t.u.
Cheval vapeur × .9863 = Horse Power.
(Centigrade × 1.8) + 32 = degree Fahrenheit.
Franc × .193 = Dollars.
Gravity Paris = 980.94 centimeters per sec. = 32.17 feet per sec.

TABLE IV

The Equivalents of Ounces, per Square Inch, in Inches of Height of Columns of Water and Mercury

27.71 inches of water and 2.04 inches of mercury equal one pound per square inch at atmospheric pressure and 62° F. Temperature. Mercury is 13.58 times as heavy as water.

Ounces.	Inches of Water.	Inches of Mercury.	Ounces.	Inches of Water.	Inches of Mercury.
.146	0.25	.018	7	12.12	.892
.292	0.51	.037	8	13.85	1.019
.438	0.76	.055	9	15.59	1.148
.584	1.01	.074	10	17.32	1.275
1	1.73	.127	11	19.05	1.402
2	3.46	.255	12	20.78	1.529
3	5.20	.382	13	22.52	1.658
4	6.93	.510	14	24.25	1.785
5	8.66	.637	15	25.98	1.913
6	10.39	.765	16	27.71	2.036

INDEX